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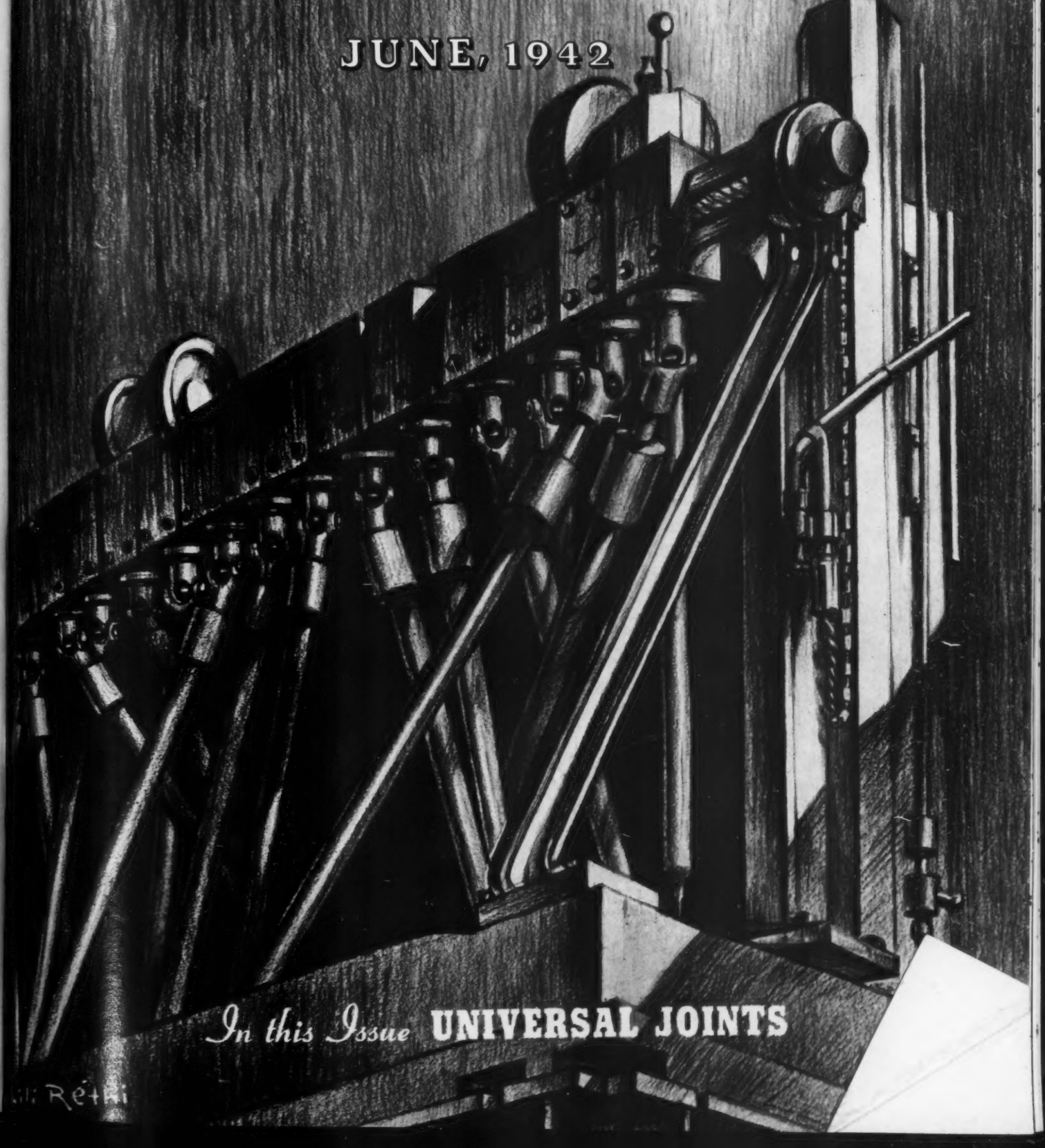
PRODUCT ENGINEERING

JUNE, 1942



In this Issue **UNIVERSAL JOINTS**

Helpful "Staff Work"

IN THE BATTLE OF PRODUCTION

All over America, engineers and production men are converting plants to war needs. These men know they can... upon request... get helpful suggestions from us about ways to speed up the working of metals and alloys.

This helpful information has been mobilized into convenient charts, pamphlets and other printed pieces. These tools-in-type range from technical data for engineers to simplified reports on performance of alloys containing Nickel under specific operating conditions.

Here, also, are correlated latest reports from the field about selection, fabrication and uses of ferrous and non-ferrous Nickel Alloys.

And, as further support for your battle of production, we offer personal assistance from members of our technical staff. With recent experience in many plants facing materials difficulties, these men become especially helpful. During wartime, Nickel... and information about Nickel... must go where it best serves the United Nations.

☆ ☆ *Nickel*

THE INTERNATIONAL NICKEL COMPANY, INC. 67 WALL STREET NEW YORK, N. Y.

PRODUCT ENGINEERING

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IN THIS ISSUE

Post-War Prospects

Although it may be too early to develop in detail designs for post-war products, it is none too early to give thought to what the conditions will be when the war is over. We know now that whole price structure of raw materials will be radically different. We know now that significant strides have been and are being made in the further development of processes new and old, such as powder metallurgy, induction heating, welding and metal forming. New design techniques are also coming into use as evidenced by all-steel and wood-veneer airplanes. Post War Prospects—this month's lead article.

Gear Tooth Profiles

To produce spur gears capable of transmitting large loads at high speeds with quiet operation has required extensive research, experiment and study. Some of the best of this research has been done in aircraft industries, and beginning on page 328 of this issue Charles G. Pfeffer, gear engineer of Wright Aeronautical Corporation, brings PRODUCT ENGINEERING readers up to date. After a brief résumé of the earlier forms of gear tooth systems, the author describes some of the new modifications.

Power Thrust Linkages

Straight-line power devices are standard units that can be applied in many types of machines to perform numerous different types of moving jobs. The design "kink" lies in the hook-up of the power unit to the machine element to be moved. Sketches on pages 326 and 327 show 21 different ways to do it, including toggles, pulleys, levers, cams, gears.

Sleeve Bearings—II

Continuing on page 322 of this number the discussion of the factors that govern the design and performance of sleeve bearings, Part II of "Sleeve Bear-

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ings," begins with the reasons why the maintenance of bearing clearance under conditions that cause deflection is important. This is followed with an analysis of the effect of surface quality. The author also developes the subject of selection of materials for bearings. The article is supplemented by graphs on two Reference Book Sheets which indicate relation between various factors used in bearing design. See page 365 for these data. Authority is of the best: Bruno Sachs, technical director of Johnson Bronze Company.

Universal Joints

On page 334 begins an article that shows how bearing loads and torques in universal joints can be determined by a simple vectorial analysis—simple compared to the tedious method of calculating loads at equilibrium in each joint and member. The vectorial method is the work of L. D. Hagenbook and J. H. Holstein, engineers with Goodman Manufacturing Company.

What Type of Plastics?

When a molded plastic part can be made by either compression or injection molding, the design engineer uses the answers to 11 common questions in making his choice. Among these questions are: Why are tool costs less for injection molds? Can cavities be blocked off? Are compression and injection molds interchangeable? Why are large parts cheaper when made by compression molding? D. A. Dearle, author of the book *Plastics Molding*, gives the answers to these questions, beginning on page 331.

Hydraulics in Road Building

Neatest application of hydraulics we've seen for a long time is a unit that mounts on the back of a road-building tractor. It pumps oil through hoses into jacks on equipment such as scrapers and earth movers towed by the tractor, and thus controls the road-building machines. Ever see a tractor-powered bulldozer rip a tree out by the roots? If so, you have a clear picture of the requirements of this hydraulic system. Dust. Vibration. Tough shock loads. The operator works fast and needs quick-operating controls. How Gar Wood Industries met these requirements is described fully in "Modern Designs," page 312.

Electric Strain Gages

A relatively new and simple method for indicating the strain in engineering materials under load makes it possible for engineers to measure exactly under almost all conceivable conditions, the loads, stresses and strains in mechanisms and structures. Description is given this month on page 337. Significance: data so obtained will promote more rational and efficient design, for such knowledge will eliminate much of the guesswork often involved in stress calculations. Author: Frank Mittelberger, mechanical engineer with printing machinery makers R. Hoe & Company.

Rectangular Shafts

Merhyle F. Spotts, of Northwestern Technological Institute, writes for PRODUCT ENGINEERING this month an article showing how to determine stresses and deformations in rectangular shafts subjected to twisting moments. Spotts' analytical procedure, beginning on page 319, is somewhat different from the usual methods. After a brief review of the concept of membrane analogy, the author applies the principles set forth to an analysis of the conditions existing in a rectangular bar under torsion.

Special Slide Rules

Slide rules to solve frequently used single equations or formulas are so easy to make that their construction should be no obstacle to their use. Though serving the same purpose as monographs, they are more rugged and often are simpler to operate. On pages 340 to 342 Ralph S. Diserens shows how to make them.

Materials Section

Substitutes for pearlitic manganese steels are suggested on pages 343 and 344. These steels contain 0.5 to 2.0 percent manganese, and are not to be confused with austenitic steels containing 10 to 16 percent manganese.

Too-extensive substitution of vanadium steels for nickel steels prompted the Advisory Committee on Metals and Minerals to ask its Metals Conservation and Substitution Group to report on where vanadium must be used and where other metals will serve. Pages 345 and 346.

Hycar synthetic rubber, of the modified Buna class, has a number of good industrial qualities: resistance to oils, solvents, sunlight and oxidation. Data on page 347.

WHAT'S COMING

Aircraft are Different

The whole world is wondering how effectively the great American automobile industry will be able to produce military planes. Much has been written about it; there is much that cannot be publicized at this time. But the huge job of conversion has already gone far enough to establish as a fact that, with the guidance of the aircraft engineers, the automotive industry will boost U. S. airplane production to staggering figures. The most interesting question is "What are the most important fundamental differences between automobile and military aircraft production?" Answers will be given in the lead article in July PRODUCT ENGINEERING.

Springback in Flanging

In the October and November 1941 numbers of PRODUCT ENGINEERING, R. G. Sturm and B. J. Fletcher of Aluminum Company of America Research Laboratories showed how to calculate springback when forming aluminum members. In July, F. B. Chapman, T. H. Hazlett, and Wm. Schroeder of the Research Department of Lockheed Aircraft Corporation will show how to compensate for springback when flanging hard aluminum sheets.

Measuring Angles in Space

Piping layouts as in ships, aircraft and machinery, usually involve various angles in different planes. Using conventional methods to determine the angles is a slow and laborious job with many chances of error. Next month Herbert Stevens, consulting engineer, will present a new graphical method.

High-Strength Plastics

The plastics industry is just beginning to get down to brass tacks and bring out the strength data needed by engineers designing structural parts. Good step in this direction is an article to appear in PRODUCT ENGINEERING for July. Prepared by H. R. Moyer of Westinghouse Electric, the article will present, by means of easy-to-read charts, structural and physical data covering four each of typical Westinghouse laminated and molded compounds.

PRODUCT ENGINEERING

GEORGE F. NORDENHOLT, *Editor*

Facts Combat Prejudice

Though often misled by prejudice and passion, he was emphatically an honest man. —Macaulay

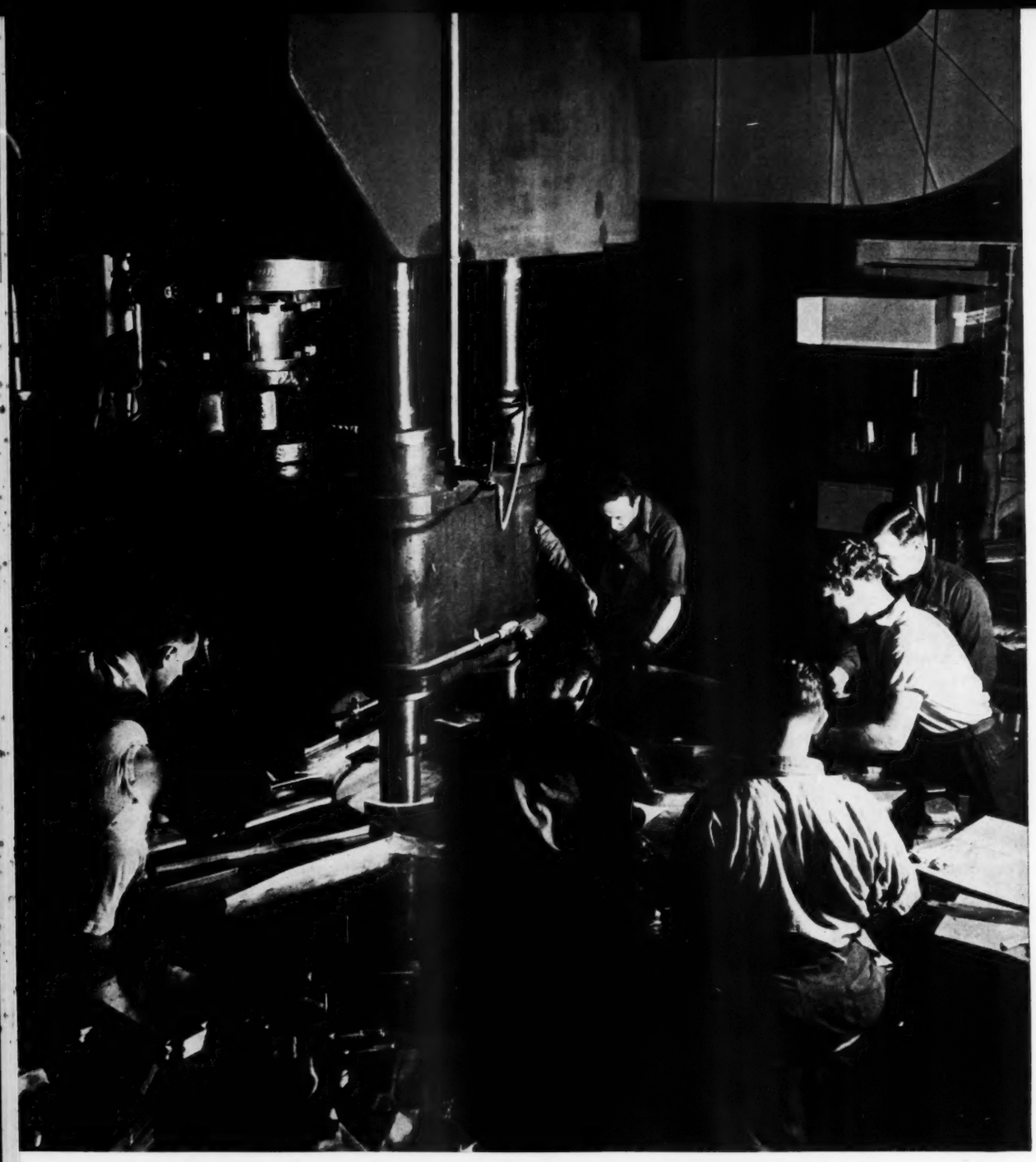
WE MUST NOT LOOK WITH TOO MUCH SCORN upon the United States senator who was quoted as saying that airplane manufacturers should be compelled to adopt modern methods such as steel dies and to discard antiquated wooden, rubber and zinc die equipment. Of course, he showed a lack of complete knowledge but of greater significance is the probability that his reasoning went something like this: "The ancients used wooden forms on which to beat out copper vessels, therefore wooden die blocks must be out-of-date; in the mass production of automobiles, steel dies are used because steel is hard and strong while wood and zinc are comparatively weak; when airplanes were being made in small lots it was alright to use wooden dies but now that the government is ordering planes in large lots, the plane builders should be compelled to adopt the modern mass production methods of the automobile builders and discard the ancient methods."

The lesson to be learned from such false logic is that it is not confined to congressmen; it is representative of a line of reasoning that is only too easy to fall into. Because of their training, engineers and scientists are much less likely to make the all too common mistake of jumping to conclusions before all the facts have been established, but this does not mean that engineers are not vulnerable to the kind of false reasoning exhibited by the congressman.

Particularly at a time like the present, engineers must guard against making snap decisions concerning the possible uses of the innumerable new materials, production processes and design techniques that are now being made available. Often their usefulness seems limited to the field for which they were specifically developed but final judgment should be withheld until after a close study has revealed all the facts. The new development might be the solution to a major problem with which the designer is concerned.

Advances in engineering are being announced so rapidly that it is difficult to keep fully abreast. However, it is an axiom that success goes to those who constantly increase their fund of knowledge, who view each new development as a possibility for further improvement and who enthusiastically sweat to reach their goal. Such men now have a golden opportunity to prove their worth, but they must get their proposals accepted and here again prejudice plays an important role.

In the development of a new idea, the designer arrives at his goal by the logical use of established information and the executives to whom he is presenting the new technique or design will be strongly influenced to arrive at the same conclusion if the facts are correct, are presented clearly, and in a sound logical manner. Otherwise an unfavorable decision, based on prejudice rather than on the merits of the case, will probably result. Only through a clear understanding and presentation of all the facts involved can prejudice, both in our own minds and in the minds of the people with whom we are dealing be overcome.



How Lockheed stamps 20,000 pieces in eight hours on a single-action 250-ton hydraulic press using the Guerin process. The rotary table, designed by Lockheed engineers, rotates a fraction of a revolution with each cycle of the press, the finished pieces being taken off at one station and new blanks being placed in position at the next station while blanks are being formed on the press. The Presswood die blocks cost little to make or alter

POST-WAR PROSPECTS

In the Light of

NOTHING is worth the price of war but it would be foolish indeed to ignore the fact that war is the greatest accelerator of scientific progress. In fact, such progress in this present conflict is greater and more significant than in World War I or in previous wars. This is logically so because the fund of scientific knowledge today is greater than ever and, secondly, war has become so highly mechanized that victory or defeat will be determined largely by the scientific and industrial output of the respective combatants.

Every nation at war considers no price too great to pay for victory and the cheapest cost factor is the time, knowledge and material expended in

Present Developments

G. F. NORDENHOLT
Editor, Product Engineering

the war effort. No nation at war hedges on the labor and material required to build ships, tanks, guns and planes in the fastest time possible. In such a period, regardless of economic considerations, existing plants and machinery are scrapped and replaced with new plants that are more suitable for faster production. To assure maximum output, new machinery and new equipment that is more efficient are installed. By the same token, more than enough men are often put to each task. Although this is wasteful of man-hours, it is warranted if the extra number of men on the job will result in greater production.

Quantity production is useless, however, unless the war machinery produced is of higher military value and effectiveness than that of the enemy. Designs must be the best obtainable and must be improved constantly. For this reason, the cost of engineering design as expressed either in dollars or man-hours, is given scant consideration. The sole objective is armaments superior and more than enough of them, regardless of dollar costs or wasted effort.

The goal is all that counts! That is why research activities and experimental engineering projects are limited only by the man-power available to put to the task. If a new idea that might improve fighting equipment gives only the barest promise of success, research scientists and design engineers are assigned to the task of developing that idea, and they do not work to either a budget or a time clock. The cost is not a factor and if the project is successful, there is no hesitancy in scrapping equipment that has been obsoleted when that appears to be the desirable thing to do. We in the United States are in a very enviable position in that we have a tremendous manpower for undertaking industrial research and technological developments. It is because of our great educational system, our colleges and universities and especially the great research laboratories built up by industry that the United States is now able to bring to bear on the scientific and technological problems, a veritable army of splendidly trained men with finely equipped research laboratories at their command. Other nations can't match it.

At a time like this it is not permissible to be specific, but here are a few figures that give a faint idea of the tremendous manpower that is being expended on the development and design of war equipment. In his talk before a meeting of the American Society of Mechanical Engineers, J. L. Atwood, executive vice-president of North American Aviation, Inc., speaking for his industry said: "Now there are actually more designers working on improvements (in aircraft) than were required to produce the original design!" He did not say how many more engineers were being employed but one can judge that by the fact that on an average, five of the leading aircraft manufacturers increased their engineering department each in the ratio of about five to one. One of these companies that had an engineering department of about 300 men three years ago now has 1,500 men —exclusive of clerks, stenographers and secretaries—and whereas three years ago the engineers were working 40 to 45 hours per week, they are now working 60 hours per week and more. A small plane manufacturer who in 1939 had an engineering department of 30 odd men now has almost 200. One of the largest companies has a school where 200 engineers are being trained.

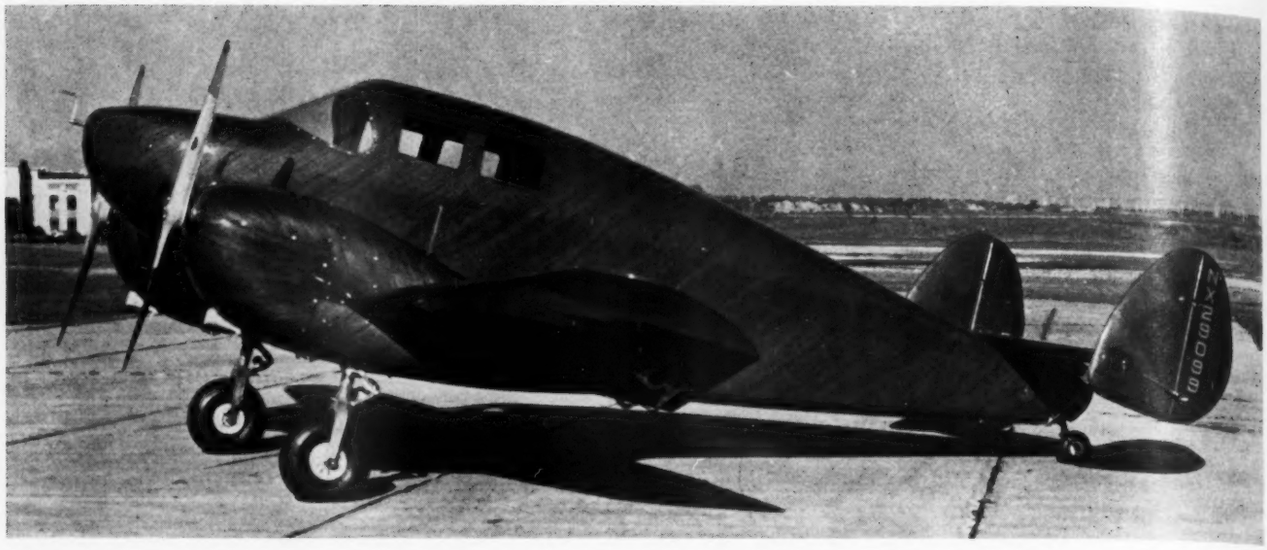
As a highly conservative estimate, and it is nothing more than an intelligent guess, more than 12,000 engineers are now engaged in further improving the design of American military planes.

These figures might appear strange in the light of the statement made shortly after the defense program began in 1940, that aircraft designs would be "frozen." In one sense they are frozen in that we are still building the same general type models as were then in production. But freezing the type models does not mean freezing the designs. When you see the symbol B-17E that means that it is the fifth improved model of the B-17 type of ship. And the B-17E is relatively as greatly different from the B-17 as a 1941 Buick 6-passenger sedan is from a 1938 Buick of the same model. Airplane engineers are in general agreement that any military plane is obsoleted in a year and a half. They hope to cut it down to a year.

But there is one great difference between the successive automobile models and the successive airplane models. In the past, the automobile companies have invariably shut down production for several months in order to tool-up for the new model. In the development of new military planes, however, the design improvements are developed in such a manner that there is practically no stoppage of production. To accomplish this apparent miracle requires highly ingenious engineering design and perhaps a great deal more of it than would be required if it were permissible to stop production. It also requires that the production methods used in the aircraft industry must be so flexible that design changes can be made without necessitating new tooling that would require a long time to make and the services of highly skilled toolmakers.

Engineering activity in the aircraft industry is typical of that in all major plants engaged in manufacturing war equipment. In one of the leading automobile plants converted to the manufacture of machine guns and cannons, the engineering department was doubled. Another company that went into the manufacturing of tanks organized a whole new engineering department. It soon grew to be three times bigger than their before-the-war organization. Even relatively small companies engaged in the manufacturing of accessories have found it necessary to expand

High-frequency induction heating developed primarily for forging and heat-treating is now being used in the processing of laminated veneers to furnish the heat to thermo-set the bonding resins while the sheets are held under pressure. Here is a Tocco machine heating the ends of shell forgings for the nosing—in operation at the S. A. Woods Machine Company plant



Resin-bonded wood veneer planes have emerged from the realms of possibilities into the fields of military use as indicated by this Langley all-plywood plane. The new design and production techniques which first had to be mastered will revolutionize many fields of design in the post-war period, ranging from furniture and architecture to airplanes and other transportation equipment

their engineering department from a handful of men to several dozens.

Unless one has studied the operation of an engineering department in a war industry, it may be hard to explain what all these engineers are doing. Most of their activities are war secrets but the mention of some of their accomplishments which have been released for publicity will give some idea of what is going on.

When the M-3 tank was designed, its shortcomings were obvious. Even months before it was put into production, metallurgists and design engineers started on the development of a new model. First, it was necessary for the metallurgists to develop a new cast armor. This was done by the best metallurgists of six independent foundries working together. Then it was necessary to develop a foundry technique to cast and heat-treat the huge tank hull. The tank then had to be redesigned completely in all details. An idea of the magnitude of this job is gained from the fact that a complete set of blueprints and specifications weighed almost 900 pounds. But in addition to this, new production methods, new plant layout, new inspection procedures and all the other problems of manufacture had to be solved by the engineers.

The availability of new materials always holds forth great promise for design improvements. In the aircraft field, hundreds of engineers are giving their full time to the application of plastics, especially the laminated sheets and the resin-bonded wood veneers. At least three different companies have developed complete designs and have built test models of trainer planes made entirely of resin-bonded wood veneer. Planes built of the same material are being developed for active military services, details of which are confidential. North American Aviation, Inc. has developed a plane built with common carbon steel sheets and resin-bonded veneer. In fact, developments in the technique of structural design indicate that it is possible to design a plane to be made entirely of carbon steel sheets and weighing only 3 percent more than an aluminum plane of comparable size.

One of the bottlenecks in aircraft engine production is the casting of aluminum cylinder heads. To produce the cast fins requires the cores and molds to be reinforced with thousands of steel needles, a slow and tedious hand operation. The cylinder head is now forged by an entirely new process. The availability of this process made it necessary not only to redesign the cylinder head but also to make numerous other changes in the design of the engine, thereby creating a new model that is more powerful, more efficient and much easier to manufacture.

Recently, one of the oil companies announced a new fuel with a considerably high octane rating. This means a redesign of all engines to take advantage of this development, but the new engines will be more powerful and more efficient. Aircraft engines of 3,000 hp. and even higher are on the horizon.

In the field of hydraulics, systems to operate on 2,000 lb. per sq.in. pressure are being tested and developed and systems operating on 3,000 lb. per sq.in. pressure are being considered. New designs of electric motors developed primarily for aircraft indicate sweeping changes to come in motor applications in all fields. Numerous new instruments for electrical communications are now in military use and many electronic devices that before the war were considered merely toys, have been perfected for uses unsuspected by laymen.

Under the pressure of war, chemists are bringing forth new materials at an outstanding rate, especially in the field of plastics. New production methods and improvements in old ones are being perfected and put into use at a dizzy rate. The use of Presswood die blocks and rubber pads for forming sheets is by no means a crude production method. On the contrary, it is highly efficient, economical and satisfactory. It will have a far-reaching effect as it makes possible small and medium lot production of sheet metal stampings at low cost, the tool costs being negligible.

For glueing wood veneers, cold setting resin glues have been developed and the wood veneers are being glued under pressure by putting them in a rubber bag and evacuating the rubber bag. If heat is required, the bag with the part in it is put in an autoclave and subjected to steam under pressure. Veneers are also being glued in hydraulic presses with heat furnished by a high frequency current by a method similar to that used in diathermy treatment. The same principle is being applied widely for heat-treating and forging metals.

These are only a few of the things that are happening now or that have already happened and that will radically change the industrial picture from what it was in 1940. Economically, the changing industrial picture will shake the very foundations of many of our old notions. We are already experiencing significant changes in the rela-

tive prices of metals. With aluminum capacity increased from 238 million lb. in 1938 to more than 2 billion lb. by 1943, and with at least three independent companies producing aluminum, the price of it will drop considerably from the 30 cents a pound price of 1938. Even today, the price of aluminum ingot has dropped to 15 cents per pound, and a price of 10 cents per pound after the war is not unreasonable to expect.

Magnesium production, 6½ million lb. in 1938, 33 million lb. in 1941, now at the rate of 350 million lb. per year, is scheduled to go to 750 million lb. in 1943, a production of more than 100 to 1 over 1938. With volume production, the cost of magnesium is being reduced continuously.

In the field of plastics, new compounds are being developed at an astonishing rate. These include phenolic resin compounds of high impact strength, a new plastic that is extruded into tubes to replace brass and copper, and new synthetic rubber-like materials that give promise of being able to compete in price with natural rubber. Every few months, a new or highly improved plastic material is being announced.

Only a few of the relatively new developments in industry have been mentioned but another factor to be considered is the tremendous educational program forced upon industry by the necessities of conversion. As Mr. Cole, vice-president in charge of engineering of the Studebaker Corporation, put it, "We are learning from our sub-contracts for aircraft engines how to work to tenths of thousands just as readily as we have been working to thousandths. When this war is over, the 1942 model automobile engine will be so obsolete that American automobilists will be ashamed to own one. Also, the post-war engine will be so much more efficient that it will be too expensive to run a 1942 model."

Furniture manufacturers are converting to the manufacture of laminated veneer airplane parts. In their new activities, their engineers will learn much about precision manufacture and structural design as applied in aircraft. It is safe to prophesy that much of what they are now learning will be embodied in the furniture to be manufactured in the post-war period.

Much the same will apply to the design of equipment for the home, industrial machinery and, in fact, the design of almost any product.

Regardless of all the disadvantages and disruptions involved, great personal benefits to designers are bound to result from the program of industrial conversion. Engineers whose horizons have been limited by specialization now find themselves compelled to master design problems in new fields and to learn about the peculiarities of materials which they formerly ignored. Many designers are also getting a better appreciation of the fact that fundamental design principles have universal application—the design of a girder is fundamentally the same whether it be for a bridge, an airplane, a railroad car or construction machinery.

Undoubtedly another effect of the conversion program will be many changes in the kinds of products manufactured by various companies. Some companies will undoubtedly remain permanently in the general field to which they converted in the war program. Some of the companies that converted to the manufacture of aircraft accessories will remain in that field indefinitely. It is rumored that some of the automobile companies will continue the manufacture of aircraft engines after the war is over. It is expected that because of the vast number of pilots being trained and the post-war program to build thousands of government owned landing fields for public use, the peacetime airplane industry will be many times greater than it was prior to the war.

Permanent conversion will be accompanied by intense competition and the design engineers will be the focal point of that competition. Business will go to the companies that have the best designed products and products which can be marketed at the most attractive prices. Designs economical to manufacture and sell, designs that take the best advantage of the new price structure in the cost of materials, designs that are developed to be manufactured by new and more economical production methods, those will be the designs that will command the post war markets. The man who can develop them will be in great demand. The design engineers who in this emergency take advantage of the present favorable conditions to broaden their horizons will be the leading designers of the post war period.

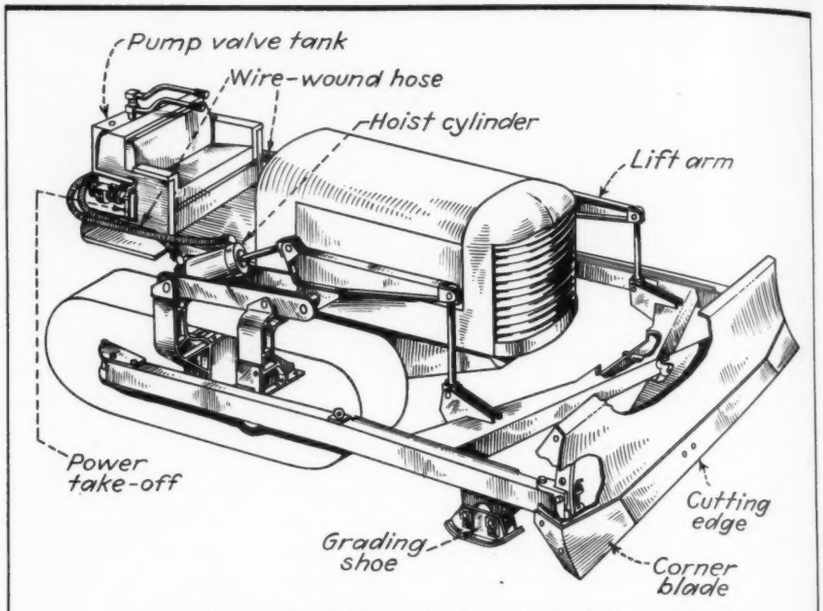
Skillfully inserting thousands of steel nails in the molds and cores for casting finned cylinder heads has been a "slow-point" in their production. New design of heads recently announced by Wright Aeronautical Corporation, made by a newly developed forging technique, will eliminate this time-consuming job and has paved the way for improved engine design

Zinc cast or Presswood die block with rubber in the upper platen of the hydraulic press, the Guerin process, is a highly efficient production method. Here is shown a stamping made in the Douglas plant. The two holes are punched out and flanged and the stamping is formed all in one operation. Die changes can be made quickly and at little cost

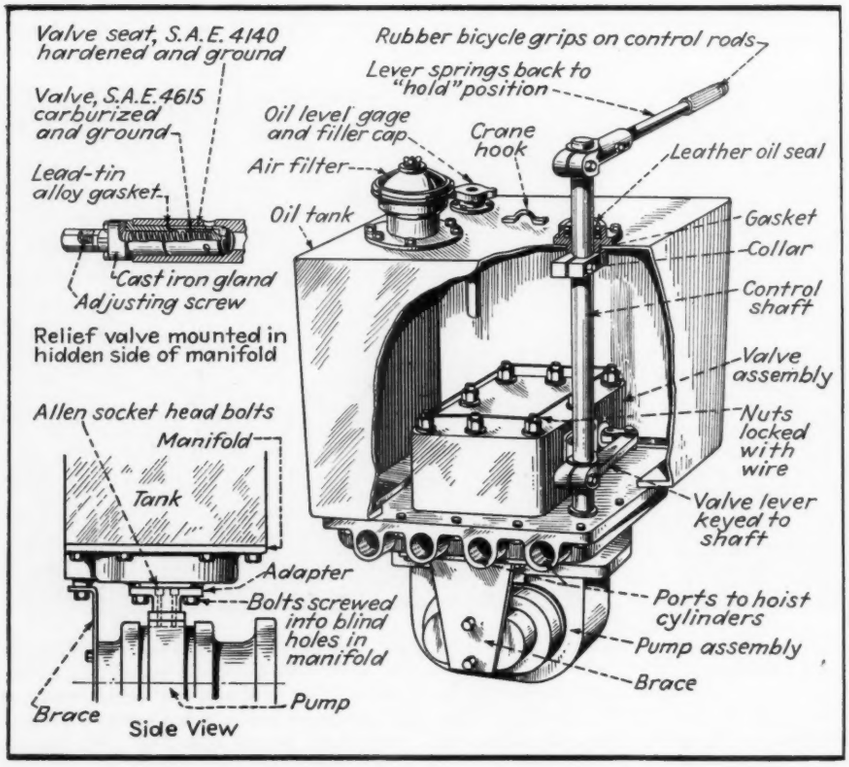
Modern Designs

Tractor-Mounted Unit Controls Earth Movers

Ideal control for raising and lowering scoops, bulldozer blades, and other elements of dirt-moving machinery would be hydraulic, reasoned Gar Wood Industries. Result, after several years of development, is this new compact unit which converts mechanical power from tractor motor into hydraulic power transmitted to hoisting jacks. Hydraulic unit gives power both up and down, is flexible so that tractor and towed unit operate independently over rough ground, prevents damage by overloading, and saves in frictional losses. Chief requirements were dust-tight, vibration-proof assembly, ruggedness, large hydraulic passages for speed, and simple design to facilitate servicing.

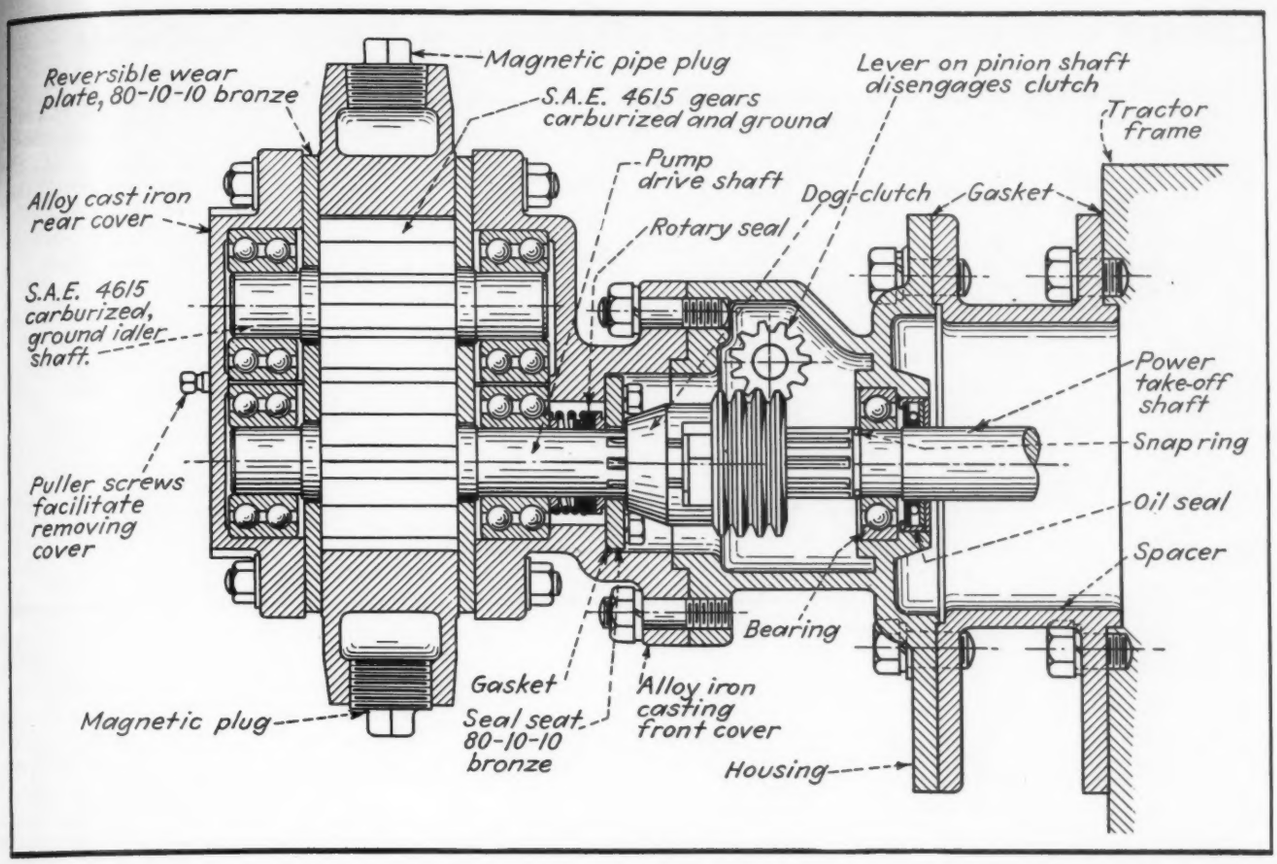


GAR WOOD HYDRAULIC UNIT ON BULLDOZER

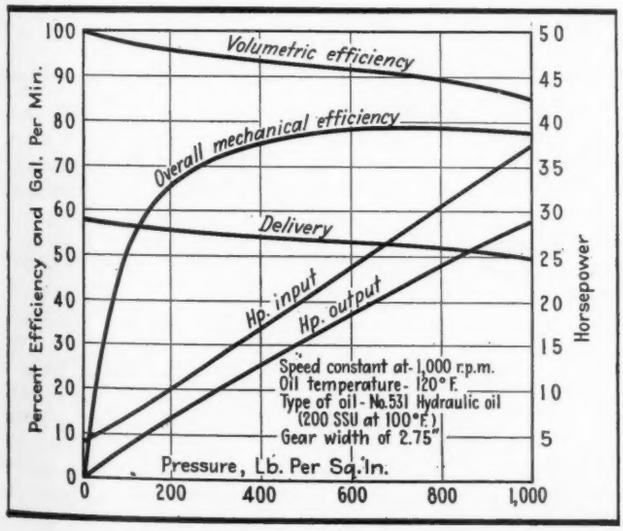


Pump, valve and tank are integral. All parts are interchangeable, and any of 3 standard pump sizes and 3 standard valves can be made up as required into units to operate bulldozers, scrapers or riggers, alone or in combination. When more than one valve is required, they are mounted one on top of another within the tank, with concentric control shafts. Simple unit shown has only one valve. By placing valve inside tank, no seals for valve spools are required, and pipe connections which might loosen under vibration are eliminated. Valve nuts are wired; other bolts have lockwashers, are closely spaced. Mating surfaces of pump, manifold and valve are machined smooth to permit use of sealing compound instead of gaskets. Hydraulic unit proper has only two oil seals: at control shaft and in pump. Control shaft seal is leather which retains lubrication although exposed to weather.

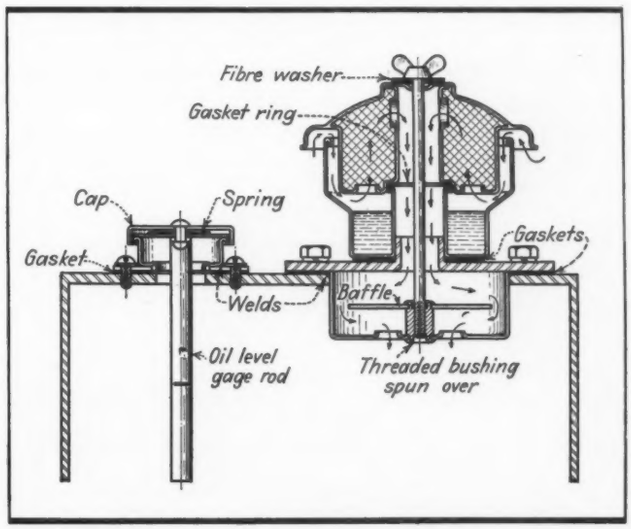
Relief valve, an easily removable unit, prevents overloading and is factory-set to relieve at 450-800 lb. per sq.in. depending on size of machine. Passages are large to prevent overheating of oil. Valve parts are hardened and ground.



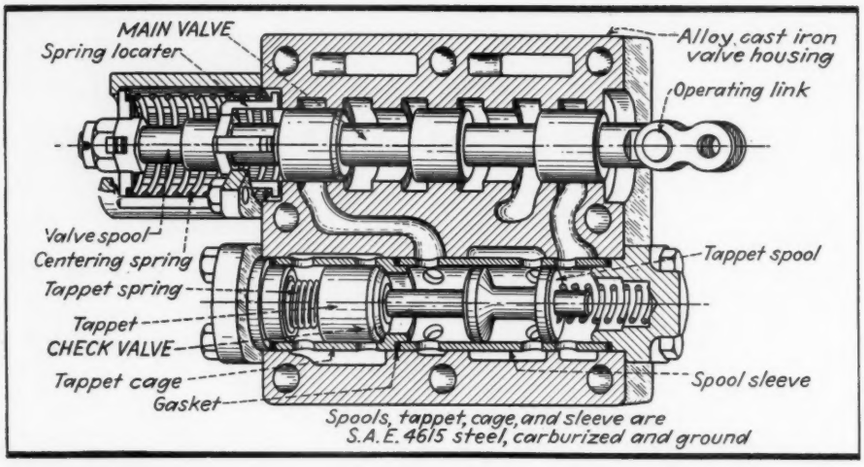
Gear type pump has molybdenum steel gears, carburized and ground. Steel shaft turns in large double-row ball bearings. Pump has reversible wear plates which can easily be turned over when one side is worn, and which can be refaced when both sides are worn. To facilitate removal of cover when overhauling, two pull-out screws are set in. Pump shaft is sealed by rotary mechanical seal. Magnetic plugs in intake ports catch all small metallic articles in oil. Pump is designed with more bolts and overall length is kept to a minimum in all three available sizes. Pump is bolted to housing of power take-off and can be removed without disturbing other parts. Principal reason for many of these details was to make it easy for the road contractor to do his own servicing.



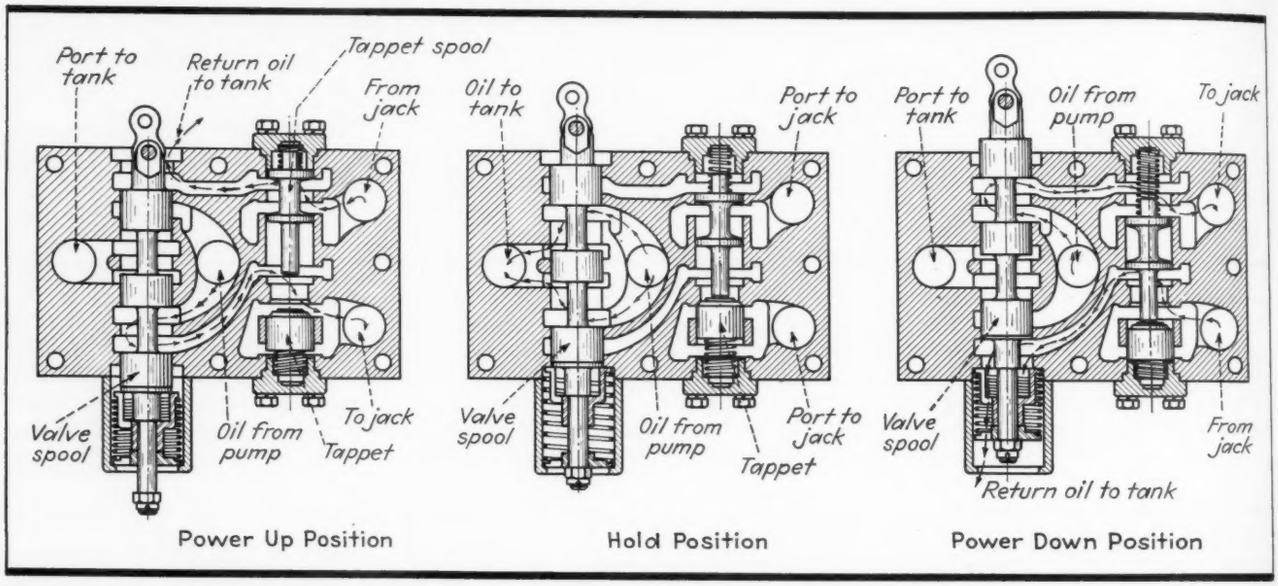
Typical performance curve of standard Gar Wood pump shows only slight drop in delivery as pressure increases to 1,000 lb. per sq. in. All oil passages in the pump are large to give high efficiency and minimum temperature rise in the oil. Oil used is S.A.E. 10 below 40 deg. F.; S.A.E. 20 between 40 and 80 deg. F.; and S.A.E. 30 above 80 deg. F. temperature.



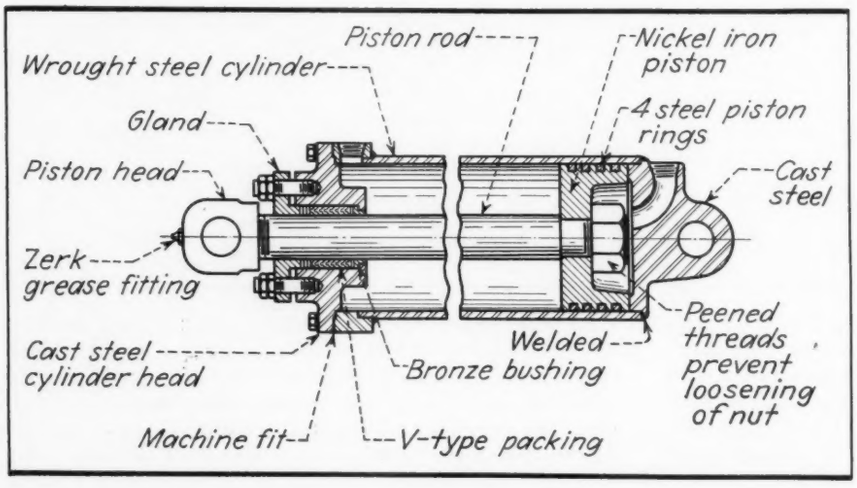
Grit and dust, always present around tractors, are kept out of oil system by this oil-bath air cleaner through which tank "breathes." Size and speed of machine, and need for relatively low oil pressures require high pump delivery; on some machines delivery is 60 gal. per min. This makes filtering the oil impracticable. *(continued on next page)*



Valve spool in Gar Wood hydraulic unit is so large that leakage caused by any slight wear might permit the blade or bowl to settle. To prevent this, an hydraulically operated check valve gives positive holding seal. In early designs, distortion of valve housing at working pressure often caused spool to stick. This was corrected by increasing wall thickness and switching to alloy with high modulus of elasticity. Built-in positioning stops and springs assure correct spool position. Check valve tappet and spool are separate pieces not affected by misalignment. Valve ports exit into tank. Any leakage also goes directly into tank. This eliminates necessity for gaskets and seals.



Operation of valve is shown in expanded diagrams above. Tappet spool mates with check valve such that return line opens to tank before power line opens to cylinder, regardless of flow direction. Control handle spring-returns to "hold" position when released. Two other standard valves, designed for use with other types of equipment, are practically the same as this, but check valves are slightly modified. They can easily be converted from one type to another.



Hoist assembly is conventional double-acting hydraulic cylinder of large diameter to give high power up or down at relatively low hydraulic pressures. Base, cylinder and piston rod are rugged to withstand high shock loads. Four steel piston rings prevent leakage and consequent settling of the blade or scoop of earth-moving equipment. Gland can easily be tightened to adjust for wear in V-type leather packing. Locknuts prevent loosening of gland under vibration. Head bolts are closely spaced. Cylinder is of welded construction. These hoists are usually mounted in pairs; one combination of road-building units may contain as many as six of them.

Butyrate Plastic Chosen for Utility Lantern

Thirteen parts of the new Focal-Lite hand lantern—practically the complete assembly except for reflector, bulbs and conductors—are molded of cellulose acetate butyrate (Tenite II). This was Focal Company's solution to materials selection and fabrication problems encountered after they had added design features to meet all needs of farmers, servicemen and sportsmen. With 5½ in. reflector and 6-volt battery the lantern throws a ½-mile beam. Quick adjustment under thumb focuses the "spot" anywhere within range or throws it out of focus for floodlighting. Lantern can be operated as code signal lamp by slight thumb pressure on switch nut when nut is ¼ turn from full "on" position. Unique feature is a drop light for reaching inaccessible places; drop light is pulled back into housing by crank type reel.

Handle is designed to fit comfortably into a man's hand swinging freely at his side, and is open enough to carry over wrist or forearm. To change the bulb, reflector assembly can be removed by springing apart the plastic arms which lock it in place. Molded plastic guards prevent damage to lens and drop light. Lens is now glass, but will be molded acrylic plastic. Battery burns 80 hr., giving at least 45 hr. of strong light.

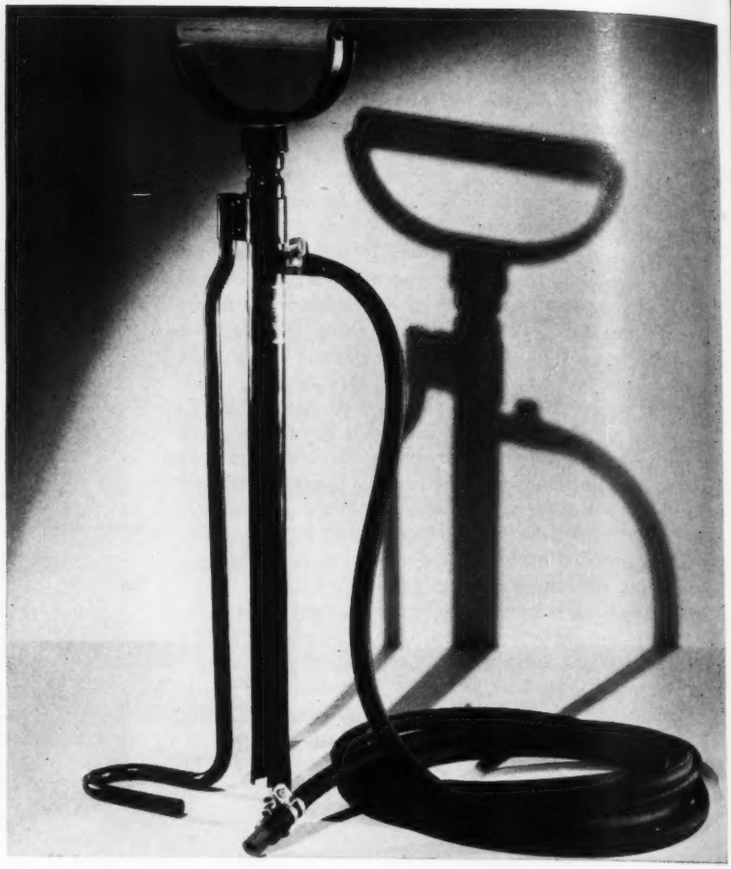
Butyrate plastic was chosen for its light weight, toughness, corrosion resistance and color. Focal Company reports weight of plastic case is only half that of aluminum case and says plastic will not dent, crack or break in rough service for which lantern is designed. Handle is assembled by screws and is mounted on battery box by threaded bushings and spherical nuts which seat in spherically recessed lugs molded on box.

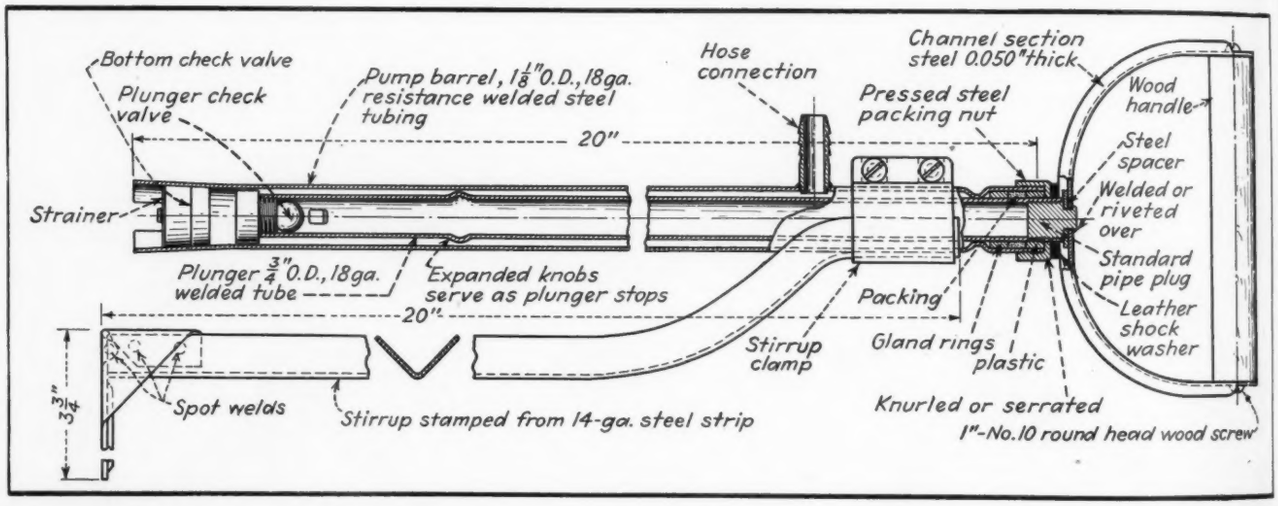
Battery box
Bottom of battery box
Battery contactors
Compartment for cable
Compartment for drop light cord
Knob twists to turn light on and off, slides to focus bulb
Drop light
Spherical nut seats in slotted lug on battery box
Cable
Cable contactor
Handle molded in halves
Threaded bushing
Molded reel
Short shafts inserted in reel
Molded recess for reflector base
Recess for drop light
Molded arm holds reflector assembly
Die-cast sliding socket tube assembled over plastic sleeve on cable
Die-cast focusing slide projects as guard for drop light

Details of lantern are illustrated above. Thin brass strips fastened inside walls of battery box carry current to pivot bushings. Current then passes into short shafts in ends of plastic reel. Circuit to focusing bulb goes from one end of reel through spring bronze contactor to sheath of flexible cable, thence through switch to insulated wire in cable, thence through bulb to die-cast socket tube, and finally to other end of reel through another spring bronze contactor. Cable sheath is insulated from socket tube by plastic sleeve. Wires of drop light cord pass through small holes in plastic reel and connect to shaft inserts, thus completing the drop light circuit.

MODERN DESIGNS – No Brass in O.C.D. Stirrup Pump

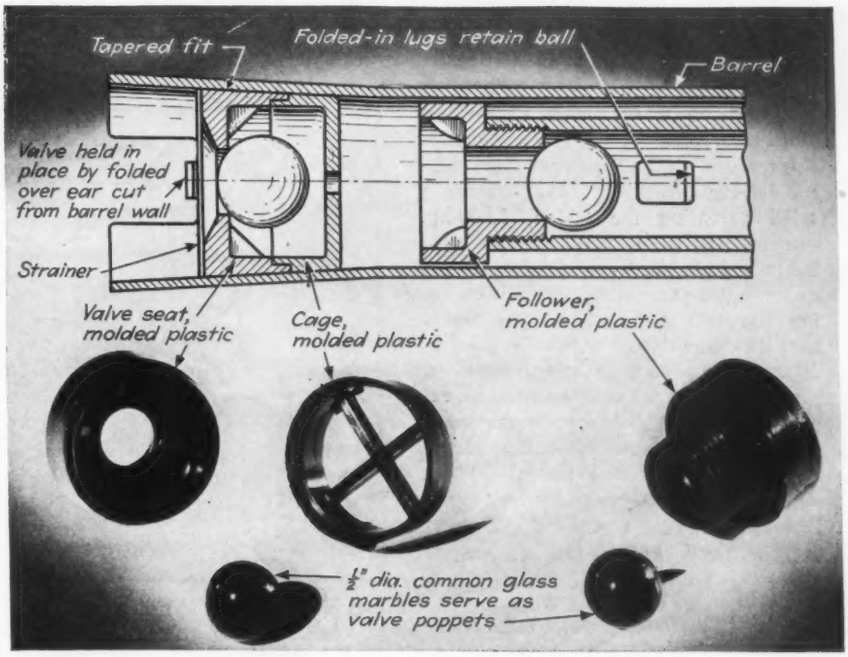
To prepare occupants of every building to combat incendiary bombs, several million bucket stirrup pumps will eventually be needed throughout the United States, O.C.D. estimates. Tough problem: stirrup pumps formerly were made of brass—now too strategic a material for this use. Also, these brass pumps cost from $8 to $23—more than the average purse can afford. What was needed was: (1) a sturdy, clog-proof, rust-resistant, easily handled pump which produces either a fine spray or steady stream of water; (2) it must be made only of relatively nonstrategic materials; (3) the design must be simple and must permit alternate fabrication methods; (4) price must be kept low. At request of O.C.D., a preliminary design for such a pump was drawn up by W.P.B. and submitted to the welded steel tube and plastics industries for refinement. Result is this just-announced stirrup pump made of welded steel tubes, simple sheet steel parts, molded plastics, glass marbles, and vinyl plastic hose. W.P.B. has established priorities for 2,000,000 of them.



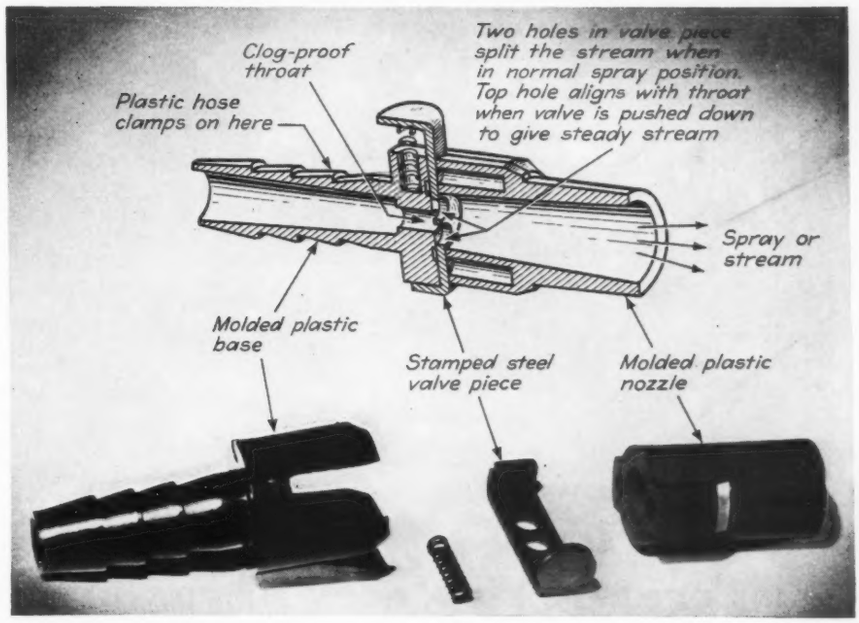


Operation of pump: On upstrokes water is drawn through intake suction valve into space below plunger. Since pressure valve in plunger is closed, water above plunger is forced out into hose and also into accumulator chamber in plunger tube. On downstroke lower suction valve closes; water in barrel below piston passes through pressure valve into plunger tube and space above piston. Air pressure in accumulator and displacement of plunger tube walls at same time force water out into hose, maintaining steady flow. Pump barrel is 1⅛ in. dia., 18-ga. welded steel tubing; internal weld fin is smoothed off. Plunger is ¾ in. dia., 18-ga. welded tube, is much stiffer than rod used on average stirrup pump. All machining operations on these tubes are simple, can be done several ways. For instance, necked-down groove near top of barrel can be formed either in beading machine or by simple lathe roll attachment. Packing nut is stamped from S.A.E. 1020 steel. Stirrup can either be formed from ½ in. dia., 16-ga. welded steel tube or from 14 ga. strip steel. Tubular stirrup is more rigid, requires only 60 percent of the steel that is required by steel stamping. Stamped alternate is offered to relieve welded steel tube industry. All steel parts are finished with rust-resistant chemical black finishes except for barrel and plunger which are coated with rust-resistant baked enamel.

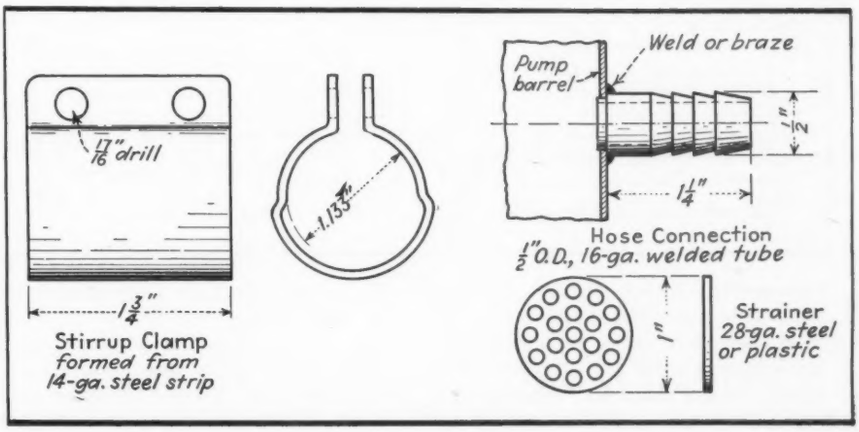
Valve seats and cage are molded cellulose acetate butyrate plastic. Valve poppets are ordinary ½-in. dia. glass marbles. Bottom check valve cage, consisting of two molded butyrate parts which together form a unit with tapered outside diameter, fits into flared bottom end of barrel and is held in place by folding in two ears stamped in the tube walls. Holes cut out by these lugs also serve as intake ports when pump barrel rests on pail bottom. Plunger check valve is another ½-in. dia. glass ball which seats on a follower screwed into the bottom of the plunger tube. Follower is a molded plastic part screwed into plunger. Clearance between follower and smooth bore of barrel is small so no cup packing is necessary. Two lugs cut out of plunger and folded in above marble prevent it from rising more than ⅜-in. in plunger. Holes stamped out for these lugs also serve as outlets for water passing from plunger to upper portion of barrel.


Tapered fit
Folded-in lugs retain ball
Barrel
Valve held in place by folded over ear cut from barrel wall
Strainer
Valve seat, molded plastic
Cage, molded plastic
Follower, molded plastic
½" dia. common glass marbles serve as valve poppets

Plastic nozzle consists of two molded cellulose acetate butyrate parts with simple stamped valve part assembled between them. Plastic parts are cemented with acetone. Valve piece slides between plastic pieces, and has two closely-spaced holes which in normal position lie half-way across the nozzle hole so as to split the stream of water. Water then rebounds from walls of carefully designed nozzle to form a spray. When steady stream is desired, thumb pressure on the valve pushes it in so that one of the holes aligns with the nozzle hole. When thumb pressure is released, valve spring returns to spray position, reducing the danger of accidentally directing a heavy stream of water on the incendiary bomb. Openings in the valve piece are large enough to prevent clogging by any dirt particles which can pass through strainer in foot of pump barrel. To simplify molding, grooves in hose connection can be machined.

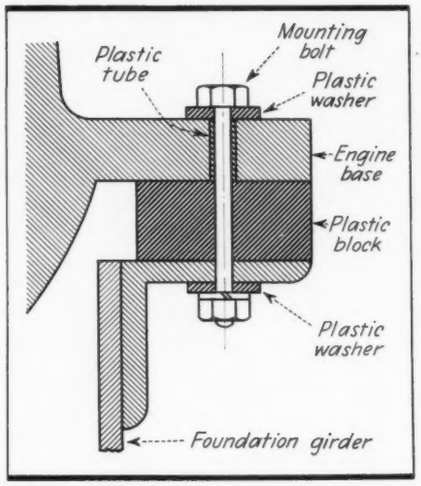

Clog-proof throat
Two holes in valve piece split the stream when in normal spray position. Top hole aligns with throat when valve is pushed down to give steady stream
Plastic hose clamps on here
Spray or stream
Molded plastic base
Stamped steel valve piece
Molded plastic nozzle

Hose connection is turned from a short piece of ½ in. dia., 16-ga. welded tube. This part can be attached to barrel by (1) brazing; (2) welding; (3) silver soldering; (4) flattening the barrel wall at joint and heading over the connection tube inside. Stirrup clamp for stamped steel stirrup is stamped from 14-ga. sheet steel. Clamp for pumps having tubular stirrup (not illustrated) is formed in one press operation from a short length of 2 in. dia., 16-ga. welded tube. Strainer can be made from sheet steel or from sheet plastics. Stamped stirrups are designed so that they can be nested for shipping when knocked-down.

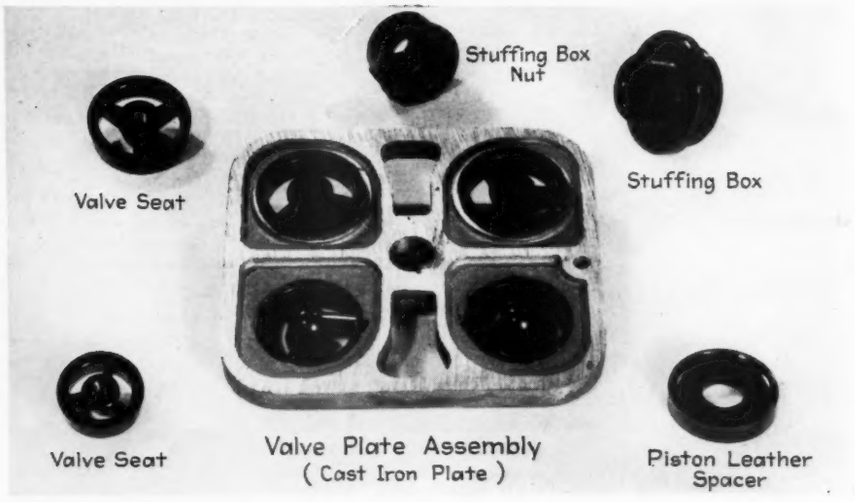


MODERN DESIGNS – Diesels Mounted on Plastic Blocks

To reduce noise and vibration from diesel engines in tugboats, Ira S. Bushey and Sons, builders of the 120-ft., 450-ton diesel tug *D. T. Sheridan* tried mounting the 60-ton, 1,000-hp. diesel engine on 40 resilient blocks of laminated plastic. So successful was the mounting in preventing noises from being transmitted throughout the vessel that the same design will be applied to 10 more tugs now building. Resilient plastic blocks are 8x6x2¾ in. in dimensions, and are made of Micarta laminated phenolic resin. Bolts are insulated by resilient laminated phenolic tubes and laminated washers under bolt heads and nuts. Nuts are assembled with lockwashers or locknuts. Two of the 40 mountings are illustrated at far right.

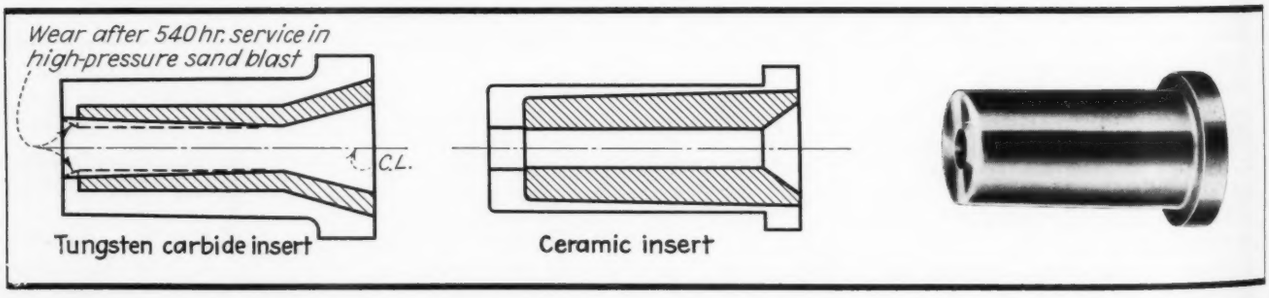


Cast Iron and Plastics Instead of Brass



Valve plate assembly of Dayton Pump and Manufacturing Company formerly was an integral brass stamping with valve seats embossed thereon. Present valve plate, as shown, is cast iron with molded phenolic resin valve seats. Cast iron valve plates are just as satisfactory as the original sheet brass valve plates from service standpoint, but cost is somewhat higher because more material and machining is required. Plastic valve seats were found superior to experimental cast brass seats although assembly and replacement required more care because of their brittleness. Other pump parts changed to molded phenolic resin were stuffing box and nut, and piston leather spacer. Shapes of these parts were not changed.

Ceramic Replaces Abrasion-Resistant Metals



Unusually hard and abrasion-resistant ceramic material is replacing tungsten carbide in inserts for American-Heanium sand blast nozzles. The ceramic is derived from alumina by a new process, and is vitreous throughout. Its properties are: good mechanical strength, high resistance to heat and impact, inertness to most chemical reagents, no water absorption after 96 hr. immersion, abrasion resistance superior to most metals commonly used, and it can be made to close manufacturing tolerances. Hardness is 9, Moh's scale (diamond is 10). Other applications under investigation include: thread guides, jewel bearings, spark plug insulators, phonograph needles, insulators.

RECTANGULAR SHAFTS

Torsional Stresses and Angular Deformations

MERHYLE F. SPOTTS
Northwestern Technological Institute

FOR BARS of other than circular cross-section the theory of torsion is complicated. For many problems solutions do not exist. While for others, for example, problems which deal with bars of rectangular cross-section, the solution for the stresses and angular deformation is obtained in the form of a convergent series.

During recent years a concept of torsion, the so-called membrane analogy, has come to the fore and has proved useful to engineers. This is an experimental method and utilizes a thin homogeneous membrane such as a soap film stretched over a hole in a plate which forms the cover of an air-tight box. The hole is made geometrically similar to the cross-section of the shaft being studied, the film across the hole is slightly bowed by air pressure and the shape of the resulting surface noted.

By mathematical analysis it can be shown that:

1. Resultant shearing stress in the bar has the same direction as the contour line of the inflated film at that point.
2. Shearing stress is proportional to the slope of the membrane at right angles to the direction of the stress at that point.
3. Torque carried by the bar is proportional to the double volume inclosed by the membrane.

Thus it is possible to solve complicated torque problems experimentally. Laboratory equipment, however, is necessary which as a rule is not available to the designer. The membrane analogy is useful, nevertheless, as it gives a mental picture of the state of stress. The engineer can imagine a film stretched across a hole in a plate similar to the shaft and can consider the places on the inflated film which would have the steepest slope, hence the greatest stress. Often improvements in design can thus be effected even though quantitative results are not available.

Stress at Corners

For example, at the bottom of a keyway like that shown in Fig. 1(*a*) there is a high stress. In fact if it were possible to make the bottom corner perfectly sharp, the tangent to the membrane would be vertical, hence infinite stress. The membrane analogy helps one to see how such a high concentration of stress can be reduced by rounding off the bottom corners. In the rectangular shaft shown in Fig. 1(*b*) there is zero shearing stress at the corners. In the inflated film, two sides of the element of the film in a corner remain horizontal, the tangents have zero slope, and hence there is no stress. For a round hollow shaft it can be proved that an additional plate representing the hole is required. This must be elevated a predetermined amount as shown in Fig. 1(*c*). It is well known that a considerable portion of the interior material may be removed from such shafts without producing much effect on the strength. This is indicated by the figure which shows that the loss of volume because of the presence of the hole is not great.

When a rectangular bar is twisted the assumptions which are applied to the case of a round bar are no longer valid. Cross-sections perpendicular to the axis of the bar are no longer plane after twisting, but are warped as shown in Fig. 2(*a*) and the mathematical determination of the state of stress is more complicated. For a rectangular bar in which the length of cross-section is great as compared to the width, the stress situation is much simplified. The membrane has contour lines as shown in Fig. 2(*b*) for such a bar.

These lines are practically parallel to each other and to the sides of the rectangle over almost all the surface. Hence, except for small regions near the ends, the shearing stress is directed parallel to the *y*-axis, and components of stress in the other directions are lacking. Also, if the volume under the film be computed by taking the cross-sectional area at the center and multiplying by the length *b* the result will be

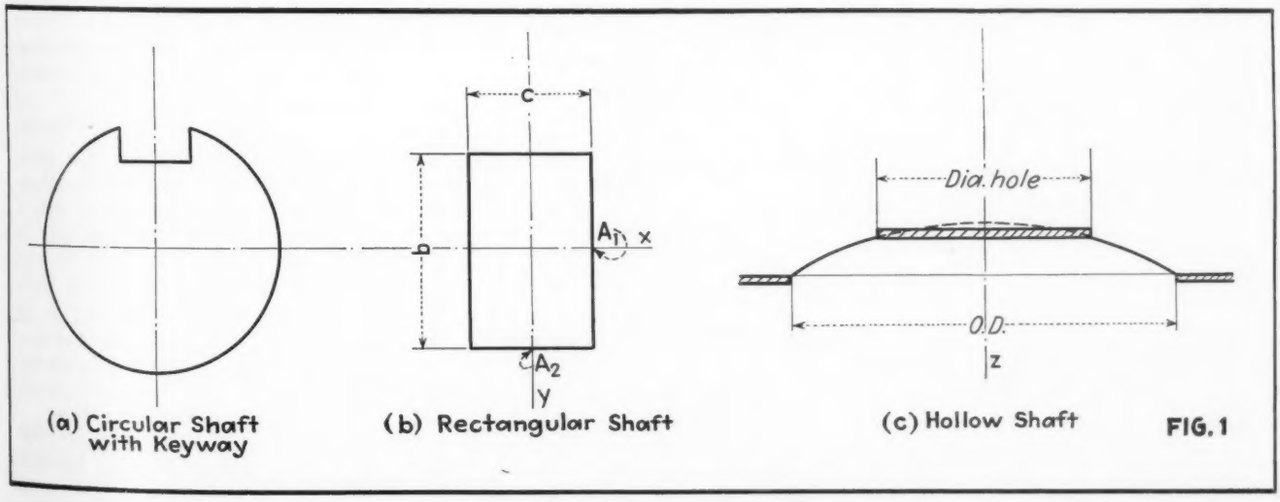


Fig. 1—Various cross-sections for shafts subjected to torsional loads

only slightly different from the actual. Based on the foregoing, an approximate solution for this problem will now be given.

Thin Bar Under Torsion

In Fig. 3(a) let the rectangular bar be twisted by couples M_t applied at the ends. It should be remembered that distance b is assumed much larger than c, and also that the length in the Z-direction is proportionally greater than shown by the figure. It is assumed that the material in the bar is homogeneous, isotropic and follows Hooke's law. Let θ_1 be the relative rotation between any two cross-sections a unit distance apart. Consider an element located at A with sides dx, dy, and dz the coordinates of which are x, y and z. Let the respective displacements of the element in the directions of the axes be u, v, and w. That is, after twisting the element has moved from point A to B having the new coordinates $x+u$, $y+v$, and $z+w$ respectively.

In the Figs. 3(b), (c) and (d) are shown three orthographic projections of Fig. 3(a), which show the displacements u, v and w of the element in moving from A to B. The element is shown oriented in its final position at B, together with the shearing stress s_{yz} which, for the problem in hand, is the only stress acting. Note that v and w are positive quantities for the element at A, but that u is negative. The cross-section in the xy-plane is assumed as being fixed against rotation.

By similar triangles in Fig. 3(b):

$$-\frac{u}{r\theta_1 z}=\frac{y}{r} \therefore u = -\theta_1 yz \qquad (1)$$

$$\frac{v}{r\theta_1 z}=\frac{x}{r} \therefore v = \theta_1 zx \qquad (2)$$

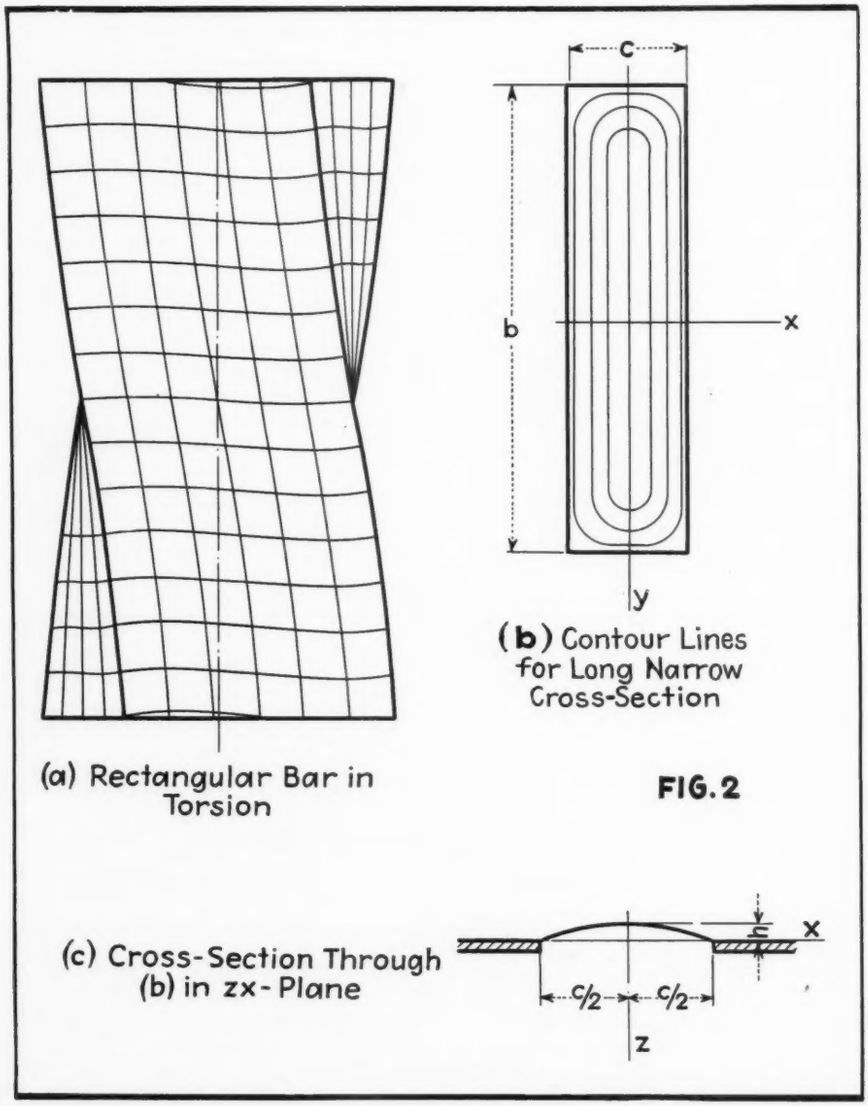


FIG. 2

Fig. 2—Thin rectangular shaft under torsion and contour lines in membrane

It is considered that the displacements u, v and w are changed by increments du, dv and dw in passing from one edge of the element to the other. For small deformations the sine may be used for the angle of shearing deformation. The total angle of shear from Fig. 3(c) then is $\partial w/\partial y$ plus $\partial v/\partial z$. By Hooke's law for shear the stress becomes

$$s_{yz} = G\left(\frac{\partial w}{\partial y}+\frac{\partial v}{\partial z}\right) \qquad (3)$$

where G is the modulus of elasticity in shear.

Differentiating Equation (2) partially with respect to z and substituting in Equation (3) gives

$$s_{yz} = \left(\frac{\partial w}{\partial y}+\theta_1 x\right) \qquad (4)$$

In Fig. 3(d) the element has no distortion in the zx-plane, and the shearing deformations add to zero.

$$s_{zx} = G\left(\frac{\partial u}{\partial z}+\frac{\partial w}{\partial x}\right)=0 \qquad (5)$$

Differentiating Equation (1) partially with respect to z and substituting in Equation (5) gives

$$-\theta_1 y+\frac{\partial w}{\partial x}=0 \qquad (6)$$

Now differentiate Equation (4) partially with respect to x and Equation (6) with respect to y and substitute

$$\frac{\partial s_{yz}}{\partial x}=2G\theta_1 \qquad (7)$$

Since s_{yz} does not vary in either the y or z directions, Equation (7) may be written as the complete differential.

$$\frac{d s_{yz}}{d x}=2G\theta_1 \qquad (8)$$

Integration gives

$$s_{yz}=2G\theta_1 x+C \qquad (9)$$

Stress s_{yz} is equal to zero when x equals zero, therefore constant C also has zero value. The final value of the shearing stress is

$$s_{yz}=2G\theta_1 x \qquad (10)$$

The stress thus varies directly with the coordinate x of the point considered as shown in Fig. 3(e). The maximum value occurs on the outer edge where x equals $c/2$ giving

$$(s_{yz})_{max.}=G\theta_1 c \qquad (11)$$

From Equation (10) and the membrane analogy it is seen that the slope of the film also varies directly with x and ordinates of the film thus vary as the second power of x. Since the tension in the film is the same in all directions the shape of the cross section, as shown in Fig. 2(c), is a second degree parabola. Values of z are given by the equation

$$z=h\left(\frac{4x^2}{c^2}-1\right) \qquad (12)$$

Differentiation gives the slope

$$\frac{d z}{d x}=\frac{8h}{c^2}(x) \qquad (13)$$

Table I—Constants for Torsion of Rectangular Shaft

b/c	1.00	1.20	1.50	1.75	2.00	2.50	3.00	4.00	5.00	6.00	8.00	10.00	∞
α_1	0.208	0.219	0.231	0.239	0.246	0.258	0.267	0.282	0.291	0.299	0.307	0.312	0.333
α_2	0.208	0.235	0.269	0.291	0.309	0.336	0.355	0.378	0.392	0.402	0.414	0.421	.....
β	0.141	0.166	0.196	0.214	0.229	0.249	0.263	0.281	0.291	0.299	0.307	0.312	0.333

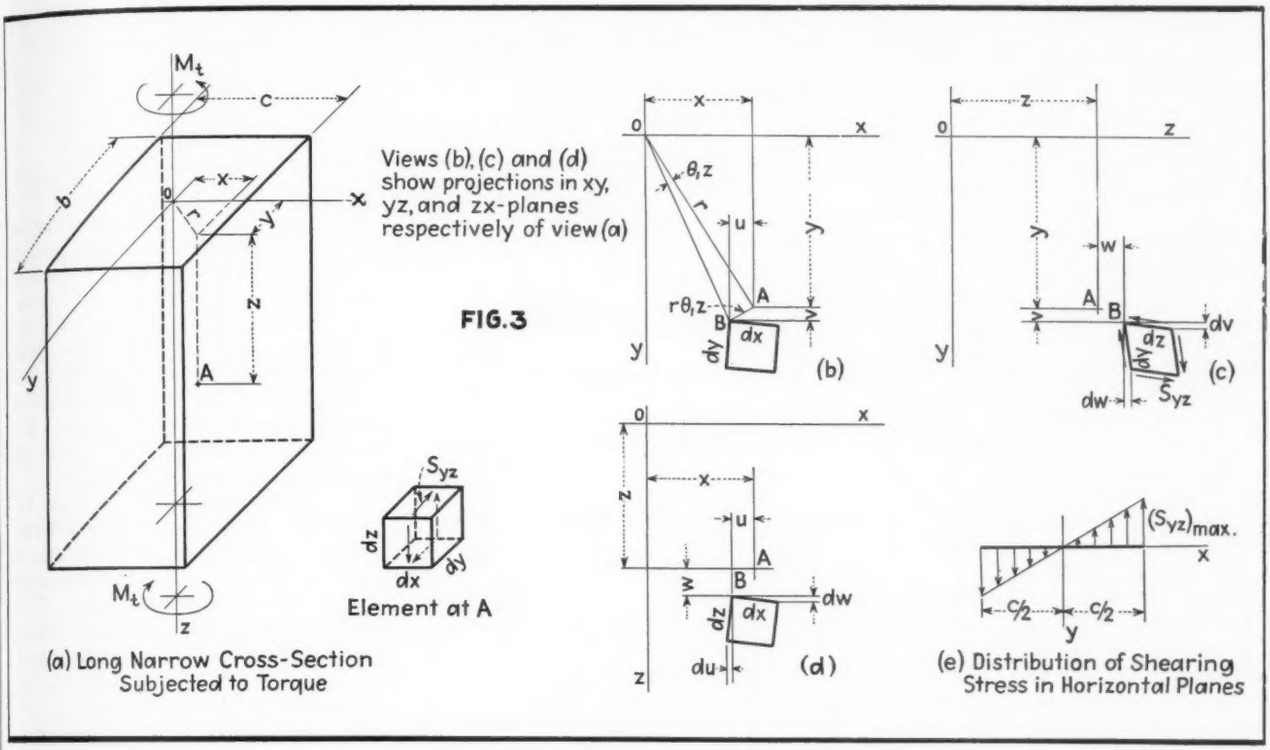

Views (b), (c) and (d) show projections in xy, yz, and zx-planes respectively of view (a)
FIG. 3
Element at A
(a) Long Narrow Cross-Section Subjected to Torque
(b)
(c)
(d)
(e) Distribution of Shearing Stress in Horizontal Planes

Fig. 3—Views showing the displacement and stresses of an element in a rectangular shaft under a torsional moment

If k is factor of proportionality, Equations (10) and (13) can be equated as

$$k \frac{8h}{c^2} x = 2 G \theta_1 x \qquad (14)$$

Whence

$$k = \frac{G \theta_1 c^2}{4h} \qquad (15)$$

Since the area under the film in the zx-plane is $2/3\ ch$, the volume inclosed by the film becomes

$$V = \frac{2 b c h}{3} \qquad (16)$$

By membrane analogy the torque is proportional to the double volume inclosed by the film.

That is

$$M_t = \frac{4 k b c h}{3} \qquad (17)$$

The factor k in Equation (17) has the same value as in Equation (14) since in the parabola of Equation (12) the slope and area are directly proportional.

Substituting for k, and solving for θ_1 gives the equation

$$\theta_1 = \frac{3 M_t}{G b c^3} \qquad (18)$$

Substituting from Equation (11) yields the value of the shearing stress

$$(s_{yz})_{max.} = \frac{3 M_t}{b c^2} \qquad (19)$$

When the width of the cross-section is relatively large as compared to its length, the foregoing derivation cannot be used. For the bar of Fig. 1(b) the general equations for stresses and deformations may be written

$$(s_z)_{max} = \frac{M_t}{\alpha_1 b c^2} \quad \text{(for point } A_1\text{)} \qquad (20)$$

$$(s_z)_{max} = \frac{M_t}{\alpha_2 b c^2} \quad \text{(for point } A_2\text{)} \qquad (21)$$

$$\theta_1 = \frac{M_t}{\beta G b c^3} \qquad (22)$$

Values of the constants α_1, α_2, and β have been computed by more advanced methods and are listed in Table I for the ratios of b/c most used in practice.

The table permits the finding of the shear stress at the mid-point of both the long and short sides, as well as the unit angular deformation for rectangular cross sections starting from a square, b/c equal to 1, to b/c equal to infinity. The maximum shearing stress occurs at the middle of the long side, using α_1. However, in cases where there are also bending stresses present it is possible, because of combined effects, that the maximum resultant stresses would occur at A_2, hence the values of α_2 given in Table I are needed.

The foregoing derivation is valid only for cross-sections not close to the ends of the member or point of application of the load. This follows from the fact that at such points the torsional moment M_t is not usually distributed over the cross-section in the manner implied by stress Equations (20) and (21), or Equation (22) for the deformation. By the principle of St. Venant one knows that such disturbances in the stress distribution are of a local nature, and that at some distance away the equations again give valid results.

SLEEVE BEARINGS – II

Factors That Govern Their Design and Performance

BRUNO SACHS
Technical Director, Johnson Bronze Company

SOME of the factors and considerations involved in the design of bearings that were discussed in Part I of this article, see May PRODUCT ENGINEERING, included heat equilibrium and bearing temperature, friction states, co-efficient of friction and types of lubrication, oil film thickness, oil viscosity and bearing clearance. Supplementing the data included in Part I, two additional charts are presented as Reference Book sheets, p. 365–366 in this issue. One chart shows relation between clearance, film thickness, coefficient of friction and other factors for bearings having a length diameter ratio equal to 1.5, the other chart shows this data for bearings having a length diameter ratio equal to 2. This second installment of the article begins with a discussion on the importance of maintaining clearance under deflection.

Under load the journal and shaft deform elastically. When in their deformed position the clearance should be sufficient so that the ends of the journal do not rub against the bearing surface. In this position some space should be present at the edges as indicated by K in Fig. 8. Since the actual clearance between the journal and bearing when aligned axially will lie between a minimum and maximum value as determined by the tolerances given for machining the shaft and bearing, these tolerances should be determined so that the end space K in the deformed position is zero when the permissible clearance is a minimum. End space K will then be provided by the tolerances existing in the shaft and bearing.

Under this condition with

D = shaft dia., in.
d = journal dia , in.
L = distance between bearings, in.
l = length of bearing, in.
a = distance of load from bearing in.
E = modulus of elasticity of shaft material, lb. per sq. in.

The minimum clearance for a load and bearing arrangement as shown in Fig. 9 is

$$c_{min} = 2.55\left(\frac{p}{E}\right)\left(\frac{l}{d}\right)^3 l + 3.4\left(\frac{p\,d\,l^2}{ED^4}\right)(2L - a)\,a \quad (9)$$

The first member in Equation (9) represents the deflection of the journal, the second member the shaft deflection.

For a load arrangement as shown in Fig. 10 the minimum clearance is

For bearing No. 1

$$c_{1\,min} = 2.55\left(\frac{p_1}{E}\right)\left(\frac{l_1}{d_1}\right)^3 l_1 + 3.4\left(\frac{p_1\,d_1\,l_1^2}{ED^4}\right)L^2 \quad (10)$$

For bearing No. 2

$$c_{2\,min} = 2.55\left(\frac{p_2}{E}\right)\left(\frac{l_2}{d_2}\right)^3 l_2 + 6.8\left(\frac{p_2\,d_2\,l_2^2}{ED^4}\right)\left(\frac{L^2\,b}{L+b}\right) \quad (11)$$

When a bearing is self-adjusting, only the first member in Equations (9), (10) and (11) has to be considered.

For applications where the shaft deflection cannot be calculated with simple statical methods, a thumb rule value for the clearance is 0.001 in. per in. of shaft dia. (c/d equals 0.001). This value can be decreased ranging down to 50 percent for babbitts and has to be increased ranging up to 50 percent for bronzes. Generally speaking, bronzes require a larger clearance than babbitts because of the deviations from ideal conditions which always do exist. Incidental metal-to-metal contacts are a consequence of these deviations and bronzes have a greater tendency towards scoring and seizing of the shaft.

The instruments now available for measuring surface roughness give an average value in root-mean-square micro-inches. The actual maximum heights of the peaks are 3 to 5 times greater, that means a surface with a reading of 10 root mean square micro-inches might have irregularities up to 50 micro-inches.

In order to have the peaks of the shaft and bearing surface irregularities separated with a film of at least the same thickness as the sum of the maximum peak heights, the total oil film thickness must be 2 times 5 times 2 or 20

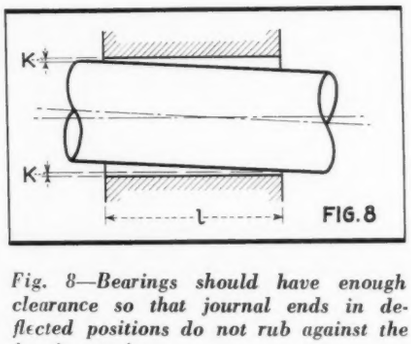


FIG. 8

Fig. 8—Bearings should have enough clearance so that journal ends in deflected positions do not rub against the bearing surface

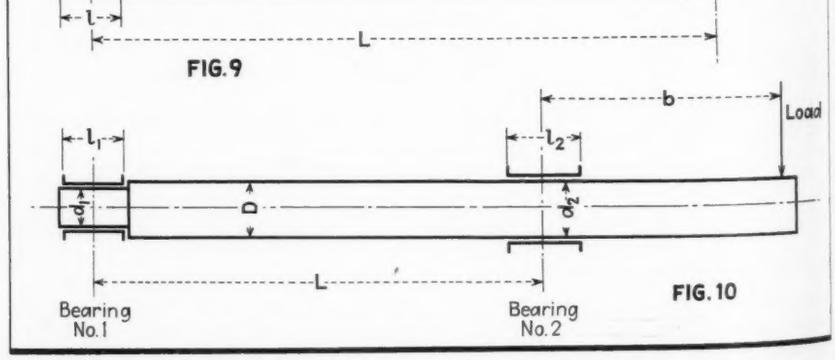


FIG. 9

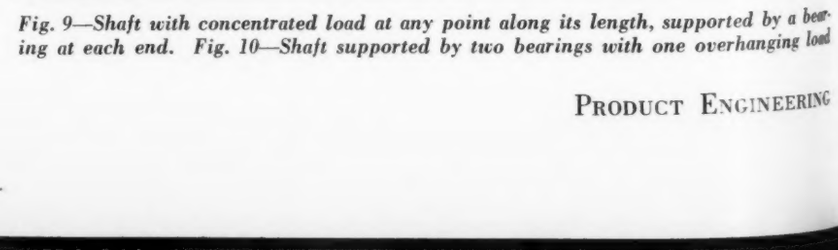


FIG. 10

Fig. 9—Shaft with concentrated load at any point along its length, supported by a bearing at each end. Fig. 10—Shaft supported by two bearings with one overhanging load

times thicker than the root mean square value of the surface roughness.

The significance of the surface finish lies in the fact that it determines the requirements for the oil film thickness; the rougher the surface, the thicker must be the film. This not only involves a higher coefficient of friction but it decreases the safety factor with which the bearing will run in the desired and calculated friction state. An example will help to understand this. Assume a journal having a diameter and length both equal to 1 in., with a clearance of 0.001 to 0.0015 in. and a film thickness of 240 micro-inches, then from Fig. 7 (May *P.E.*) the coefficient of friction f will be 0.0031 to 0.0039. The safety factor for a surface roughness of 12 is the total film thickness divided by the sum of maximum height of surface irregularities, or

$$\frac{240}{2 \times 5 \times 12} = 2$$

With a surface roughness of 3 and the same oil film thickness, the coefficient of friction will be the same as before but the safety factor will be eight.

If film thickness is reduced to 120 micro-inches, the coefficient of friction f will be reduced to 0.0024 to 0.0031, and the safety factor for a surface roughness of 3 will be four.

All of which shows that by improving the surface quality, not only the coefficient of friction can be decreased, but the bearing becomes less sensitive against being thrown out of the fluid friction state by incidental decreases of the oil film thickness caused by changes of $\sqrt{Zn/p}$ or by outside disturbances such as vibration or shaft deflection.

Furthermore, the rate of wear will become less, because the chances for incidental metal-to-metal contact are decreased; also, because during the starting and stopping periods the shaft and bearing surfaces are in direct contact for shorter periods of time.

The requirement for low surface roughness becomes more important as the bearing diameter becomes smaller, because it can be seen from Equations (7) and (7a) that the oil film thickness h is proportional to the diameter. This consideration refers to fluid as well as to semi-fluid friction.

In addition to the "regular" surface irregularities, which are detected and measured by the Profilometer, machined surfaces show also waviness or deviations from a true geometrical surface, which is defined as those surface irregularities having an axial peak-to-peak distance greater than $\frac{1}{32}$ in. Typical waviness may have widths of $\frac{1}{32}$ to 1 in. and heights of 10 to 200 micro-inches. Because of the comparatively large width of the wave, it does not respond to the measuring principle of the Profilometer, consequently it is not included in the surface-roughness-reading.

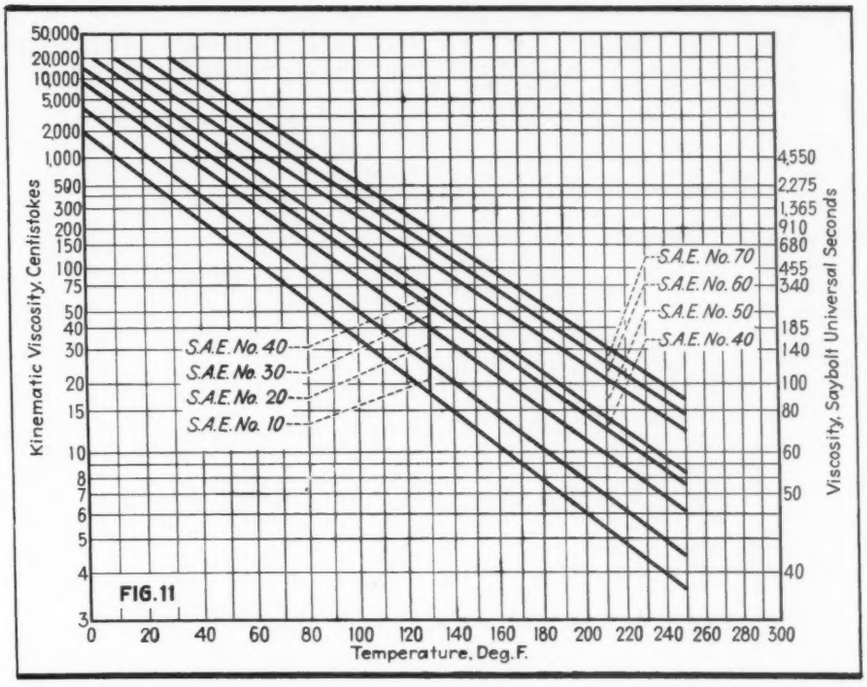


FIG. 11

Fig. 11—Straight-line chart showing approximate change in viscosity with change in temperature for typical S.A.E. number oils

Records of waviness profiles can be obtained only with specialized equipment. Profile studies of surfaces have shown that with Superfinishing methods, not only the highest degree of surface smoothness is obtained, but also the smallest amount of waviness.

Bearing surfaces obtained by good standard manufacturing methods might have waves ranging up to 100 micro-inches high and from $\frac{1}{8}$ to $\frac{1}{2}$ in. long. These deviations from the true geometrical surface together with load deflections and vibrations make a certain amount of "running-in" necessary even in present-day machines. The "running-in" operation consists of a combination of abrasive-cutting by lapping and a plastical or burnishing action.

To sum up and to illustrate the application of the foregoing principles of bearing design, the requirements for a bearing like that shown in Fig. 9 will be analyzed. Assume that bearing is to be medium ventilated and that

d = 1 in.,	l = 1 in.
p = 200 lb. per sq. in.,	
n = 1,100 r.p.m.,	D = $1\frac{1}{4}$ in.,
a = 6 in.	L = 12 in.,
Bearing temperature	= 140 deg. F.
Outside temperature	= 60 deg. F.

From Fig. 2 (May *P.E.*) the maximum permissible coefficient of friction f is seen to be 0.005. The minimum clearance $c_{min.}$ from Equation (9) is 0.001 in. This minimum clearance can be secured by a shaft dia. of 1.000 in. max. and bearing dia. of 1.001 in. min. Manufacturing tolerances of 0.0005 in. for shaft and bearing will result in a maximum clearance of 0.002 inch.

From Fig. 7 (May *P.E.*) it will be found that with

$$\frac{c}{d} = \frac{c}{1} = 0.001 \text{ to } 0.002$$

$$f = 0.005$$

$$\frac{h}{d} = \frac{h}{1} = 370 \text{ to } 280$$

$$h = 370 \text{ to } 280 \text{ micro-inches}$$

$$\sqrt{\frac{Zn}{P}} = 11.1 \text{ to } 11.7$$

The required absolute viscosity Z of the oil at 140 deg. F. then is

$$Z = \frac{11.1^2 \times 200}{1{,}100} \text{ to } \frac{11.7^2 \times 200}{1{,}100}$$

$$= 22.4 \text{ to } 24.9 \text{ centipoises}$$

Assuming that the specific gravity of the oil is 0.85, the kinematic viscosity is

$$\frac{Z}{0.85} = \frac{22.4}{0.85} \text{ to } \frac{24.9}{0.85}$$

$$= 26.3 \text{ to } 29.2 \text{ centistokes}$$

The chart in Fig. 11 shows the approximate range of viscosity change with temperature of commercial oils corresponding to S.A.E. numbers 10 to 70, however, the variations of different oils of the same S.A.E. number at temperatures other than the "definition temperatures" might be wider. From Fig. 11 it is seen that a kinematic viscosity of 26.3 to 29.2 centistokes at

140 deg. F. corresponds to S.A.E. oil No. 20 in the upper range.

With a "safety factor" against touching of the surface peaks of at least 2, the necessary surface roughness of shaft and bearing results in

$$\frac{280}{2 \times 5 \times 2} = 14 \text{ micro-inches}$$

Resuming with the results of our example, the following conditions are necessary for the correct functioning of the bearing:

A circulating lubrication system,

A shaft diameter 1.0000 in. max. to 0.9995 in. min.,

A bearing diameter 1.0015 in. max. to 1.0010 in. min.,

Surface roughness of shaft and bearing of 14 root-mean-square micro-inches,

Oil S.A.E.—20 upper range.

Now the same example will be considered with drop-feed lubrication, which results in a state of semi-fluid friction.

In order to obtain a stationary temperature of 140 deg. F. even with a well ventilated design, the coefficient of friction according to Fig. 2 (May *P.E.*) should not exceed 0.007. Such a low value cannot be obtained with boundary lubrication. The lowest values of f which can be secured with a slight degree of safety are 0.015 to 0.020. Therefore, the first conclusion to be made is that this bearing arrangement is not expedient and, if at all possible, should be changed, for instance, by increasing the journal length.

For the sake of the example no change will be made, however, a bearing temperature of 200 deg. F. will be permitted, that is, a temperature differential ΔT of 140 deg. F. (assuming 60 deg. F. room temperature), which would require a coefficient of friction of not over 0.017 (see Equation (2b) and Fig. 1, or Fig. 3, May *P.E.*). Calculations will be based on f equal to 0.02 and any excessive heat over 200 deg. F. will be dissipated by applying a stream of water on the outside of the housing.

An equation that evaluates the coefficient of semi-fluid friction is

$$f_s = 0.14 - C_3\left(\frac{Zn}{p}\right) \qquad (12)$$

where C_3 is a constant depending on operating and design conditions.

If the bearing specified in the example would be provided with sufficient quantities of oil, it would operate in a state of fluid friction down to a value of $\sqrt{Zn/p}$ equal to 7 with a value of f equal to 0.003, at which point the peaks of the surface irregularities begin to touch. These values are obtained from Fig. 7 for c/d equal to 0.0015 for the mean clearance.

This value of Zn/p equal to 49 is transposed in the diagram Fig. 12 (in which the abscissa is Zn/p and the ordinates are coefficients of semi-fluid friction) with the corresponding limit fluid-friction value of 0.003; the point obtained is connected with the point whose coordinates are Zn/p equals 0 and f_s equals 0.14. The connecting straight line represents Equation (12). The tangent of the angle β is C_3. It can be seen that the desired coefficient of friction of 0.02 is obtained with a Zn/p value of 43, or with a kinematic oil viscosity of 9.4 centistokes at 200 deg. F.

The question of the interrelation of properties of materials and their suitability for bearing purposes is quite complex. For the practical purpose of selecting a bearing metal among the existing groups of bearing metals, however, the problem can be simplified by taking into consideration these factors: Ability to run-in, to conform to "disturbances," and to prevent seizure; wear; loadability and bearing temperature, and corrosion.

The properties involved in the ability to run-in, to conform to "disturbances" and to prevent seizure are of a plastical, elastical and thermal nature. Even if a bearing was properly designed and manufactured, minute deviations from the calculated conditions as a result of mechanical imperfections and vibrations will occur. Such conditions are pronounced when the machine is new, and also during starting and stopping periods.

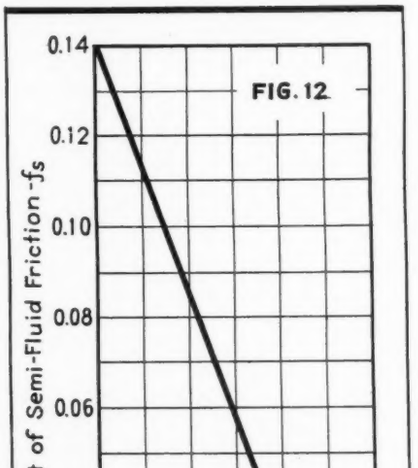


Fig. 12—Relation between "Zn/p" and coefficients of semi-fluid friction for bearings with a length diameter ratio of one

Another source of deviations from the calculated condition are the minute zones of metal to metal contact in bearings with dynamic loads.

Seizing is caused by some welding action between the shaft and bearing material. One of the necessary conditions for welding of two metals is a clean oxide-free surface for each metal in order to enable the metal atoms to come into mutual action. This surface condition might be obtained in a bearing by the scrubbing and rubbing action of the two running surfaces.

If one of the two components is present in the molten state, diffusion and welding take place quickly; if both metals are solid, a combination of elevated temperature and pressure is necessary to bring about the welding action; the lower the temperature below the melting point the higher must be the pressure to bring about the welding action in a finite amount of time.

It can be shown that high local three axial pressure states can occur in bearings, excessively higher than the linear yield compression strength of the bearing material; the magnitude of these pressures depends on the outside disturbances and on mechanical and physical constants of the material.

A measure for the ability of a metal to deal with all these deviations and disturbances is, as far as the plastical requirements are concerned, the hardness of the material at 150 deg. F. The lower the hardness figure, the more suitable is the material from the point of view of the properties considered here. Hardness figures at room temperature and at 150 deg. F. are given in Table II for various bearing alloys.

As to the elastical and thermal properties, it can be shown that the local pressures, created by local increases of temperatures (as a consequence of suppressed thermal expansion) caused by disturbances, are smaller for lower values of the expression:

$$\alpha E\left[1/\left(3 - \frac{E}{G}\right)\right]\frac{100}{g\,s}(1 - \lambda) \qquad (13)$$

wherein:

α = linear coefficient of thermal expansion
E = modulus of elasticity in compression
G = modulus of elasticity in shear
g = specific gravity
S = specific heat
λ = linear heat conductivity

Numerical values of this expression are shown in Table II for different alloys. Both the plastical and elastical values are of the same dimensions (kg./mm.^{2}) and their product is a relative measure of the ability of the mate-

Table II—Composition and Properties of Bearing Alloys

Alloy No.		Average Analysis					Aver. Brin. Hardness at Room Temp.	Relative Wear	Running-in, Conformability and Anti-disturbance Properties			Permissible Static Pressure			
J.B.	S.A.E.	Cu	Sn	Pb	Zn	Sb			Plastical Brinell Hardness at 150 deg. F.	Elastical and Thermal *Equation (13)	Total	Lb. per sq. in. at deg. F.			
												70	200	300	400
PSB or 9 Tin. Babb.	110	3.5	88.5	0.3 max.		7.5	24.5	4–6	16	26	4.2	4,500	2,000	1,500	
11 Tin. Babb.		6	87			7	25	4–6	17	26	4.4	4,500	2,000	1,500	
CT Tin. Babb.		2	89			9	22	4–6	16	26	4.2	4,500	2,000	1,500	
LX-13 Lead Babb.	13	0.25	5	84.75		10	21.5	7–11	13	27	3.5	3,500	1,500	800	
LX Lead Babb.	14	0.25	10	74.35	0.4 As	15	24	7–11	14.5	27	3.9	3,500	1,500	800	
25 BL	481	67	3.75	29			40	0.4–1	39	29	11	5,000			3,000
B4	481	70	4	26			42	0.4–1	41	30	12	5,500			3,200
Led-aloyl†		84	10	5	1% Graph.		38	0.4–1	38	40	15	6,000		5,000	
29	67	78	7	15			52	0.4–1	51	37	18	7,500			5,000
B10		70	10	20			58	0.4–1	57	35	20	7,500			5,000
66	66	85	5	9	1		50	0.4–1	49	46	23	7,800			5,500
27	64	80	10	10			61	0.4–1	60	40	24	9,700			7,000
72	660	83	7	7	3		55	0.4–1	54	55	30	8,600			6,000
71	40	85	5	5	5		50	0.4–1	49	70	34	8,200			5,500
63	63	88	10	2			66	0.4–1	64	85	54	12,000			9,000
65	65	89	11				70	0.4–1	68	85	57	13,000			9,300
51		87	10	1	2		67	0.4–1	65	88	57	12,500			9,300
53	62	88	10		2		68	0.4–1	66	88	59	13,000			9,300
55		86	12		2		74	0.4–1	72	88	63	13,600			9,300
44‡		88	4	4	4		110	0.4–1	108	102	110	20,000			14,000
40‡		89	0.5		9.5		110	0.4–1	108	110	118	20,000			14,000

* $\alpha E \left[1/\left(3 - \frac{E}{G} \right) \right] \frac{100}{g\,S} (1 - \lambda)$ † Bronze powder sintered ‡ Sheet metal

rial to run-in, to conform to disturbances, and to resist against seizing. This total value divided by 100 is also shown in Table II for the different alloys.

When considering these properties in conjunction with the selection of a bearing material, it is imperative to have in mind the safety factor of the oil film, the surface roughness, and the possible amount of disturbances. If, for instance, the manufacturing and design possibilities can not secure a certain low surface roughness and a closely controlled clearance and deflection, then babbitts are to be preferred on account of superior running-in, conformability and anti-disturbance properties.

On the other hand, for vital bearing applications the selection of a bronze alloy should be made in conjunction with strict requirements as to surface quality and oil film safety factor; closest assembly tolerances of the whole aggregate must be observed.

Wear is usually defined as continuous removal of material from the contacting surfaces of bodies, moving relatively to each other, as a result of abrasion and corrosion. Wear is not a physical property depending only upon the material or materials involved, but the result of co-acting of many additional factors such as whether lubricated or not, kind of lubricant, atmosphere, surface condition, load and how applied.

Wear tests which imitate working conditions of bearings are of practical value and from these it has been observed that the relative wear of the three groups of bearing materials ranges from 0.4 to 1 for bronzes, 4 to 6 for tin base babbitts, and 7 to 11 for lead base babbitts.

It is generally desirable to get the unavoidable wear in the bearing and as little as possible in the journal. This is obtained automatically with unhardened journals running in babbitts. With bronze bearings, the shaft wear will increase as the anti-disturbance value of the bronze increases. The advisability or even the necessity to harden the journal surface will depend on the individual application. Only the general statement can be made that the frequency and intensity of disturbances, the safety factor of the oil film, and the surface roughness have to be balanced with the anti-disturbance properties of the bearing alloy.

POWER THRUST LINKAGES AN

POWERED STRAIGHT LINE MOTION over short distances is applicable to many types of machines or devices for performing specialized services. These motions can be produced by a steam, pneumatic or hydraulic cylinder, or by a self-contained electric powered unit such as the General Electric Thrustors shown herewith. These Thrustors may be actuated manually by pushbuttons, or automatically by mechanical devices or the photo-electric relay as with a door opener. These illustrations will suggest many other arrangements.

Fig. 1—Transfer motion to distant point.

Fig. 2—Double throw by momentary applications.

Fig. 3—Trammel plate divides effort and changes directions of motion.

Fig. 4—Constant thrust toggle. Pressure is same at all points of throw.

Fig. 5—Multiplying motion, 6 to 1, might be used for screen shift.

Fig. 6—Bell crank and toggle may be applied in embossing press, extruder or die-caster.

Fig. 7—Horizontal pull used for clay pigeon traps, hopper trips, and sliding elements with spring or counterweighted return.

Fig. 8—Shipper rod for multiple and distant operation as series of valves.

Fig. 9—Door opener. Upthrust of helical racks rotates gear and arm.

Fig. 10—Accelerated motion by shape of cam such as on a forging hammer.

Fig. 11—Intermittent lift as applied to lifting pipe from well.

Fig. 12—Straight-line motion multiplied by pinion and racks.

Fig. 13—Rotary motion with cylindrical cam. Operates gate on conveyor belt.

Fig. 14—Thrust motions and "dwells" regulated by cam.

Fig. 15—Four positive positions with two Thrustors.

Fig. 16—Toggle increasing thrust at right angle.

Fig. 17—Horizontal straight-line motion as applied to a door opener.

Fig. 18—Thrusts in three directions with two Thrustors.

Fig. 19—Fast rotary motion using step screw and nut.

Fig. 20—Intermittent rotary motion. Operated by successive pushing of operating button, either manually or automatically.

Fig. 21—Powerful rotary motion with worm driven by rack and pinion.

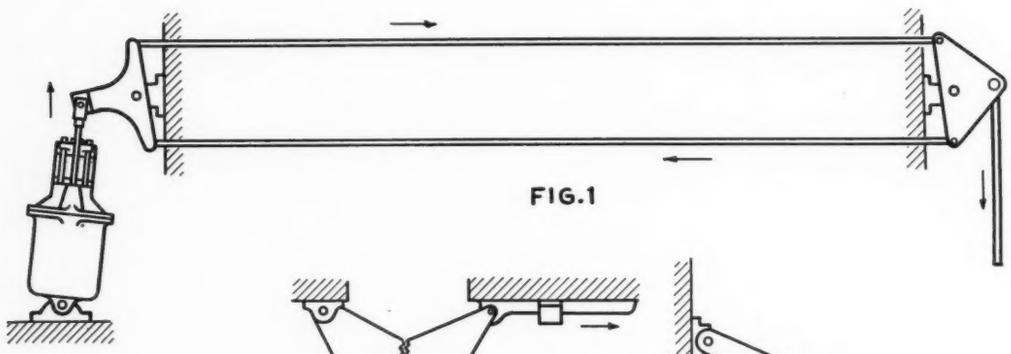


FIG.1

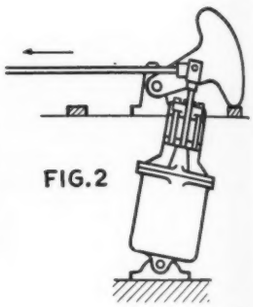


FIG.2

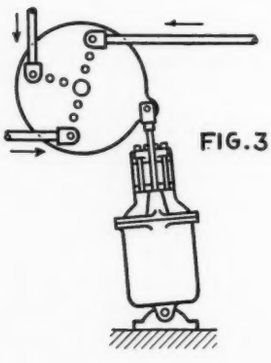


FIG.3

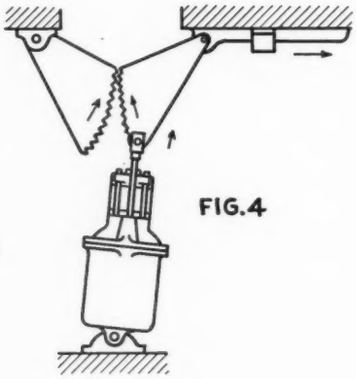


FIG.4

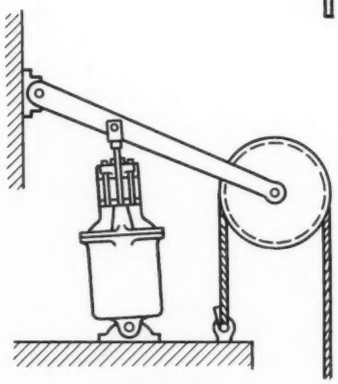


FIG.5

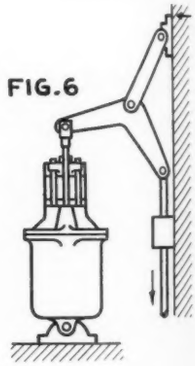


FIG.6

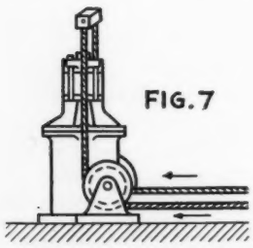


FIG.7

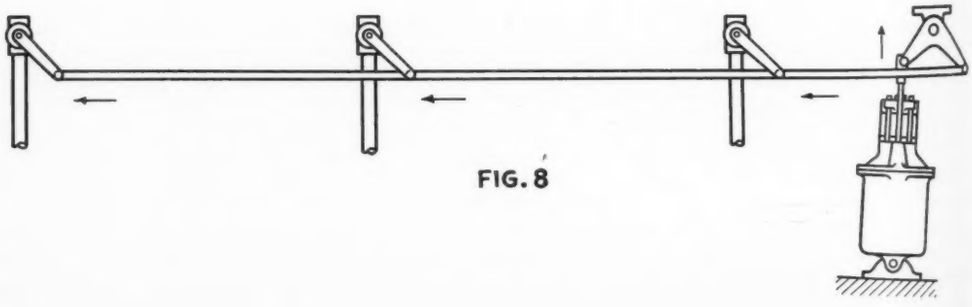


FIG.8

AND THEIR APPLICATIONS

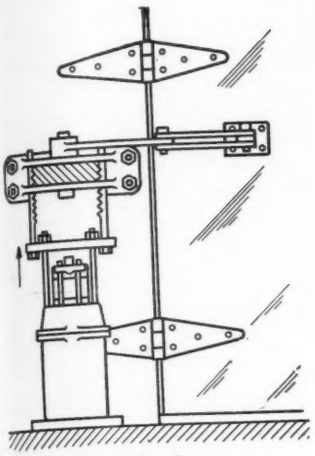


FIG. 9

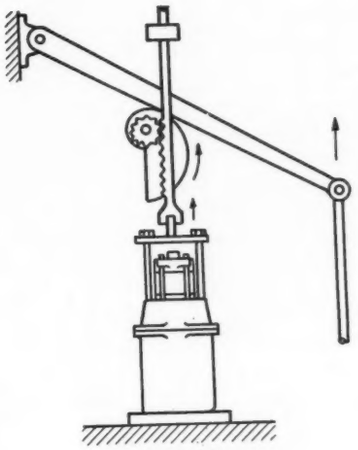


FIG. 10

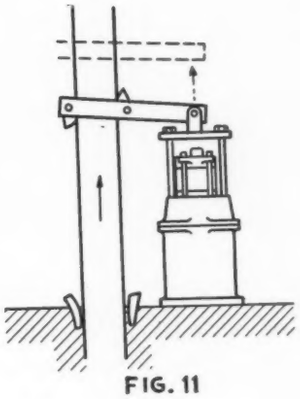


FIG. 11

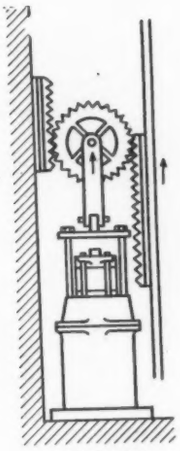


FIG. 12

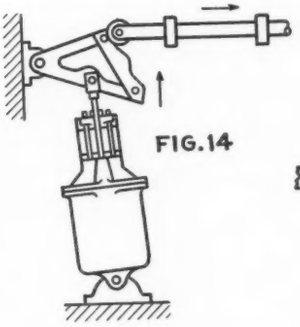


FIG. 14

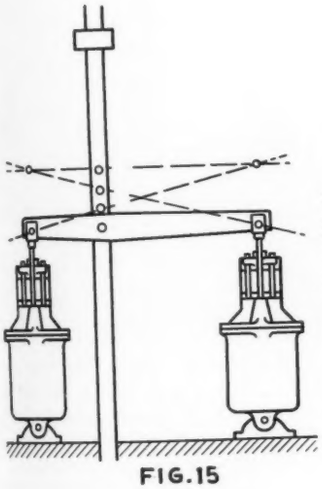


FIG. 15

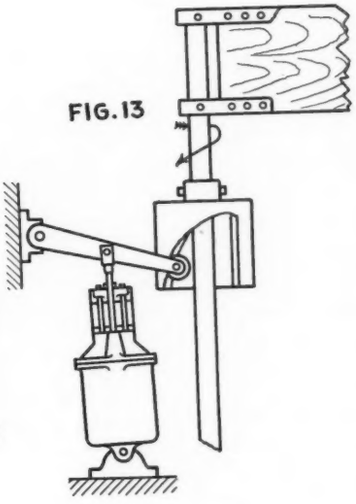


FIG. 13

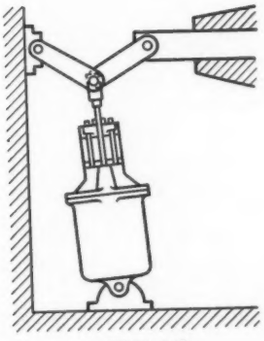


FIG. 16

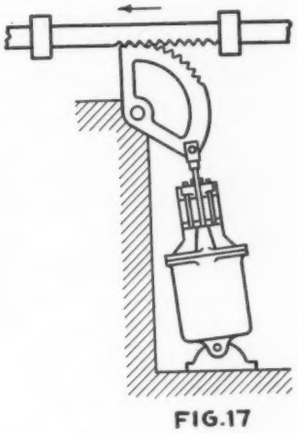


FIG. 17

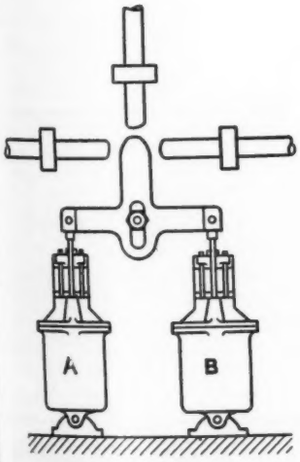


FIG. 18

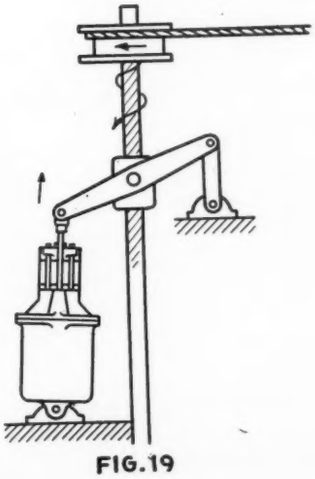


FIG. 19

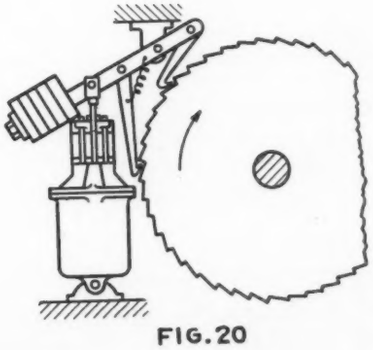


FIG. 20

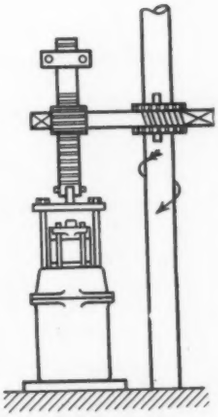


FIG. 21

GEAR TOOTH PROFILES

Factors in the Design of Gear Teeth

CHARLES G. PFEFFER
Gear Engineer, Wright Aeronautical Corporation

DEMAND for quality gears in recent years has resulted in trends and developments in gear tooth design and manufacturing methods which have naturally changed the former conception of gear applications. To produce gears which run at higher speeds, greater loads and quietness of operation has required extensive research, experimenting and study to meet these requirements. The result has been a fuller and more thorough understanding of the properties of the involute curve and the flexibility of its application. Therefore, values, existing but not used previously, are now brought to the designer's attention and full use of the advantages possible are now incorporated to produce efficient gears to the exacting specifications necessary for modern gear operation.

A brief résumé of these trends may help to clarify the progress as it developed. Thus the order is as follows:

14½ deg. Standard tooth
20 deg. Stub tooth form
20 deg. Full depth form
20 deg. Long and short addendum

First the 14½ deg. standard tooth was mainly governed by the manufacturing methods available. Therefore, design was principally controlled by what pitch and tooth ratios would suit the given center distances and such possibilities as modification of tooth form as applied today were not even considered, perhaps rightly so, as gears were slow moving and made with ample strength. That gears were used and made that way was a direct result of the use of this tooth, as any gear with less than 31 teeth would be badly undercut, requiring larger pinions and a high factor of safety.

The stub tooth system was developed to give a stronger tooth and make possible gears of smaller tooth numbers to reduce space. With this system the generating method of producing gear teeth was perfected. The 20 deg. stub system does produce a strong tooth and is a useful method for gears which do not require smooth action. But for critical gear design it sacrifices many of the more important elements such as carry-over or over-lap, contact area and load distribution, properties of equal or greater importance than the element of strength alone.

The current practice is to use full depth gear teeth. This form has many advantages. It uses a considerable length of involute curve. It can produce a suitable over-lap or tooth contact to satisfy most conditions and when modification is necessary the line of action is of sufficient length to allow it. Its beam strength is slightly less than the stub tooth but the even distribution of load makes it possible to use finer pitches. The present tendency is to use more teeth and finer pitches since finer pitches do not tend to break so easily as the bending moment is also reduced.

A further development of the full depth tooth is the long and short addendum. This is useful where small pinions are necessary and where ability to carry load evenly is required.

Many factors are involved which relate to the successfully running gears which are in operation today. Loads up to 2,000 lb. per tooth are being carried on some gears 1 in. wide. A hard and fast rule as to which form of tooth should be used for a given condition is practically impossible. Method of mounting gears, shape of gear blanks, necessary loads, speeds and accuracy of gear teeth all have a direct bearing on efficient operation. The greatest difficulty with teeth is surface failure. There are a number of important elements with which the designer should be familiar.

1. Avoid short tooth contact. To produce smooth action with gears running at greater than normal speed it is necessary to have as long a tooth contact as possible. For ordinary conditions 40 to 50 percent is sufficient, but for high-speed gears 70 percent or more over-lap should be aimed at.

2. Avoid contact near the base diameter. The involute curve has its greatest errors near the base diameter as the generating arc becomes exceedingly short at that point. Therefore, tip interference may result.

3. Equalize strength. Driving pinions usually have the smaller number of teeth and therefore they take the heaviest loads and are most likely to break. To eliminate this danger as much as possible, the tooth thickness should be increased until the beam strength factor is as large as can be permitted by manufacturing methods.

4. Approach and recess angle. The recess angle should be at least equal to and preferably greater than the approach angle. This is to avoid excessive contact at or near the base diameter of

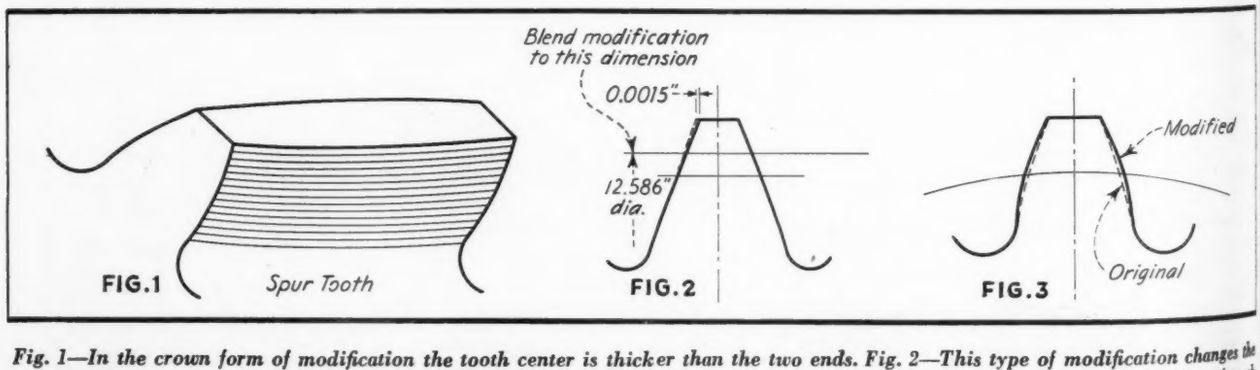
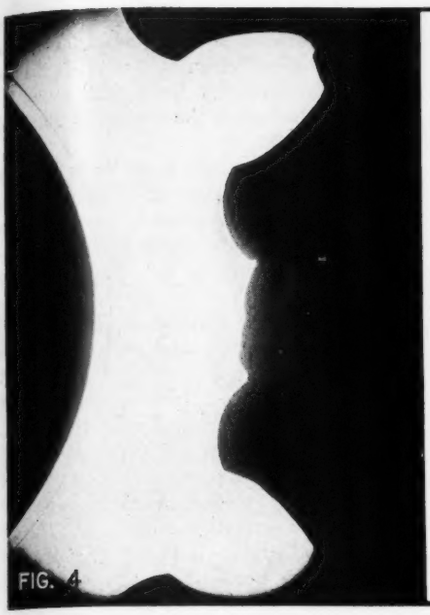
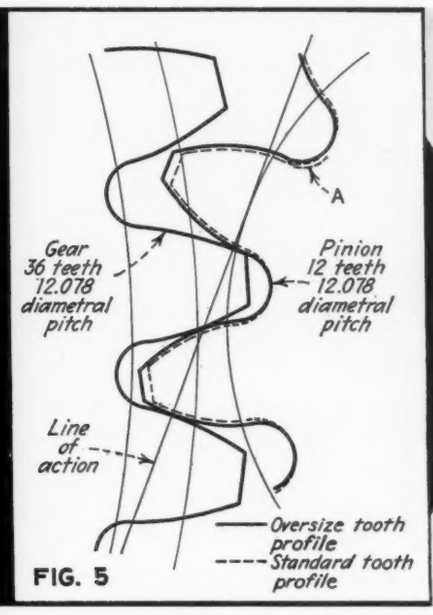


Fig. 1—In the crown form of modification the tooth center is thicker than the two ends. Fig. 2—This type of modification changes the true involute curve on one side only. Fig. 3—Barrel form tooth modification builds out working surfaces on both sides of tooth





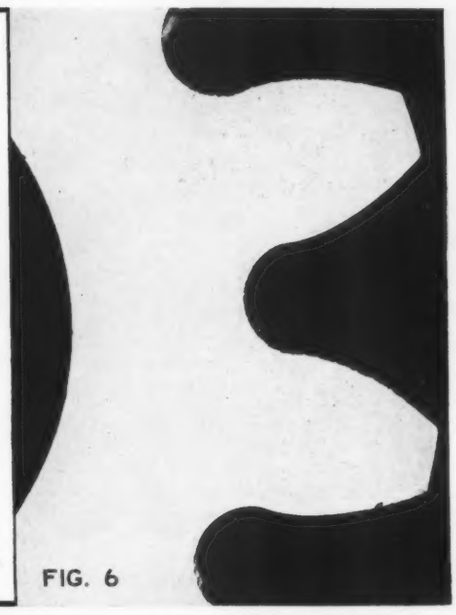


FIG. 4 FIG. 5 FIG. 6

Fig. 4—Fracture of this gear tooth began at a sharp junction of filletand tooth. Fig. 5—The radius type of root fillet is shown here. Note the oversize teeth used on pinions which are too small to roll properly otherwise. See Figs. 10 and 11 also. Fig. 6—No tooth fracture occurred when this fillet radius was used. Compare with Fig. 4 which is identical except for fillet

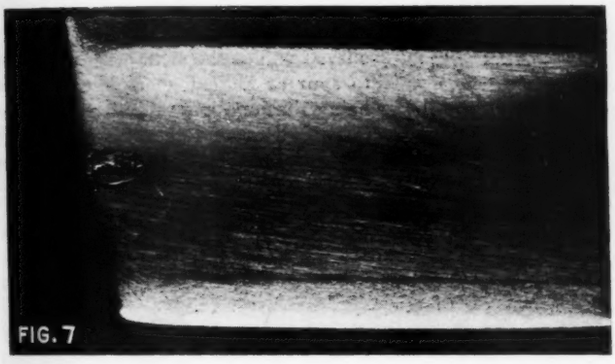


FIG. 7

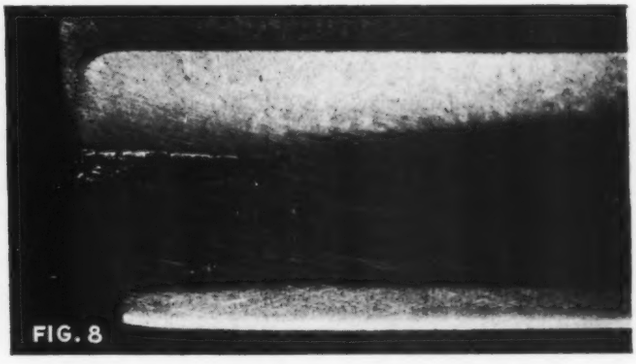


FIG. 8

Figs. 7 and 8—These two photographs show the progressive surface failure of a tooth once the surface case is fractured

the pinion and to place the tooth action nearer the pitch circle and on the addendum rather than the dedendum.

5. Modification. Various types of modification are used. Some are an advantage and are mainly necessary to compensate for tooth deflection. Modification must be used with care and study as excessive modification can readily destroy correct tooth action and contact. Deflection is present in most gear teeth and creates a certain amount of kneading and flow of surface metal. This action becomes more apparent as the teeth mesh nearer the extremities and produces undesirable bearing markings. Thus, modification at tip or flank should reduce the load pressure rather than destroy contact and should eliminate tip interference which may result from tooth deflection.

An ideal and often used form of modification is shown in Fig. 1. This is termed crown form of modification where the tooth thickness is increased slightly at the center section. Its main purpose is to prevent end bearing caused by misalignment. Obviously, the curvature must be slight as the length of tooth bearing would otherwise be reduced and localized at the center, a condition which would be objectionable under heavy loads. Another disadvantage of this type modification is that the pitch, form and pressure angle at the ends will change from the true and correct form. Where tip modification is desired it is usually introduced as shown in Fig. 2. The change here from the true involute can again be only a small amount. The third type is the barrel form shown in Fig. 3. This is a generally useful and often employed form of modification as it is ideally suited to a number of possible conditions and constitutes a desired form of tooth profile.

6. Sharp corners weaken gear teeth. As shown in Fig. 4 they are a cause of tooth fracture. Here a fatigue failure resulted from the sharp corner left in the root fillet. Fig. 5 illustrates the desired form fillet which is as near a full fillet radius as possible, blending properly with the involute form. It is not a true radius but is epicycloidal in form because it is generated by two diameters rolling together. It is a close approximation, however. An undercut fillet as shown at *A* would be more desirable than the extreme sharp corners. Fig. 6 shows the correction made in the gear which replaced the one that failed. The new gear has been giving satisfactory performance.

Pitting or surface failure is also a common result of motion and localized pressure. Usually a pitted tooth does not bear evenly across the face of the tooth. It is necessary to get good alignment first and then relieve the pitted surfaces if further adjustment is required. Figs. 7 and 8 show how the

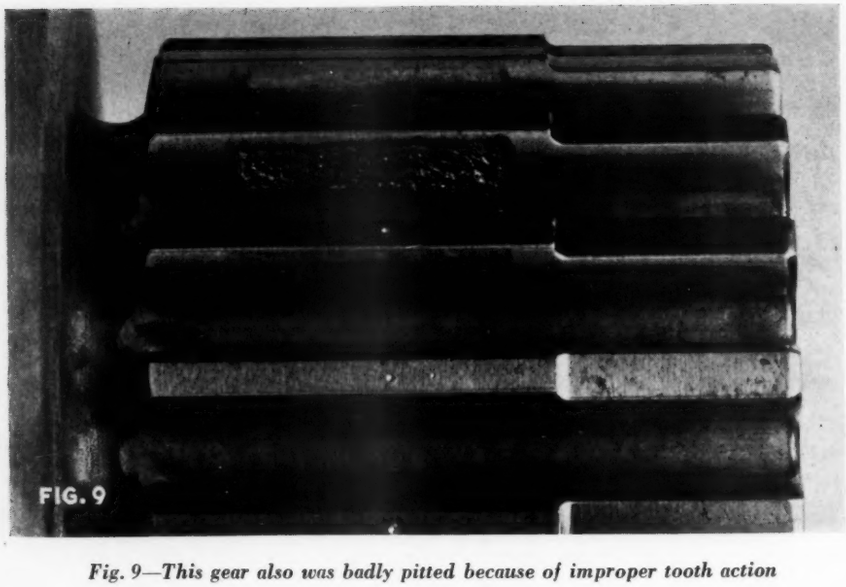


Fig. 9—This gear also was badly pitted because of improper tooth action

failure progresses once the case is fractured. Fig. 9 shows another gear badly pitted because of improper tooth action. This incorrect action may have various origins, resulting in a sliding or skidding of the surfaces which finally disintegrate and cause tooth failure.

Strut action shown in Fig. 10 is also a common cause of gear failures and noises. When the tooth is in a central position the load is carried by the tips of the two adjacent mating teeth. Further motion results in contact only at the tip and dedendum of the mating pinion, a condition obviously not conducive to efficient gearing and high load carrying ability. Fig. 11 shows the correction, the remedy being merely to increase the tooth thickness a sufficient amount to have continued contact along the line of action. This condition occurs only where small pinions are involved and is the result of high tooth ratios in which the pinion is too small on pitch diameter to roll properly. In Figs. 10 and 11 no change has been made other than that just mentioned.

A well designed and proportioned gear tooth can fail because of incorrect manufacturing procedure. For a critical and highly loaded gear, accuracy of profile, concentricity and normal pitch are just as important as good design. This is especially true when high overlap or tooth contact is present as it follows that the greater the number of teeth in contact at one time, the closer both gears must be held in correctness to each other. Thus the gear engineer is confronted with the problem of considering, not only the type of tooth to be used, but also the possibility of producing it economically and to the standard of quality necessary.

7. Strength of gear teeth. It is a well known fact that the Lewis formula is entirely inadequate for present-day gear design. This formula was developed for gears used many years ago which research and modern requirements have discarded and caused to become obsolete.

Steels of higher stress carrying ability are now available and gear manufacturing methods have changed considerably since the Lewis formula was first conceived. Many attempts have been made to establish new and more useful gear stress formulas but to date these various projects have still to be coordinated from the available data and completed.

The new A.G.M.A. series is a step in this direction. Among the studies which have been published are those in PRODUCT ENGINEERING by A. C. Rasmussen, Feb. 1939 and W. Larsen, July 1939, which investigate wear calculations as well as loading. Timoshenko's "Strength of Materials" also deals briefly with the possibility of strengthening gear teeth by the use of a full rounded root fillet. Application of this fillet has definitely shown it to be valuable and, as previously described, of great importance in eliminating one point of stress concentration. In the aircraft engine industry this form of fillet has been practically adopted as standard design and the probability is that eventually it will become universally applied to all types of gear design.

The foregoing is based on spur gear practice and no attempt has been made to show its application to helical gears. As the basic principle of the involute curve remains the same whether applied to spur or helical gears, the difference being that one is straight and the other is twisted around the axis to the helix angle, it is possible to apply the same procedure to either type.

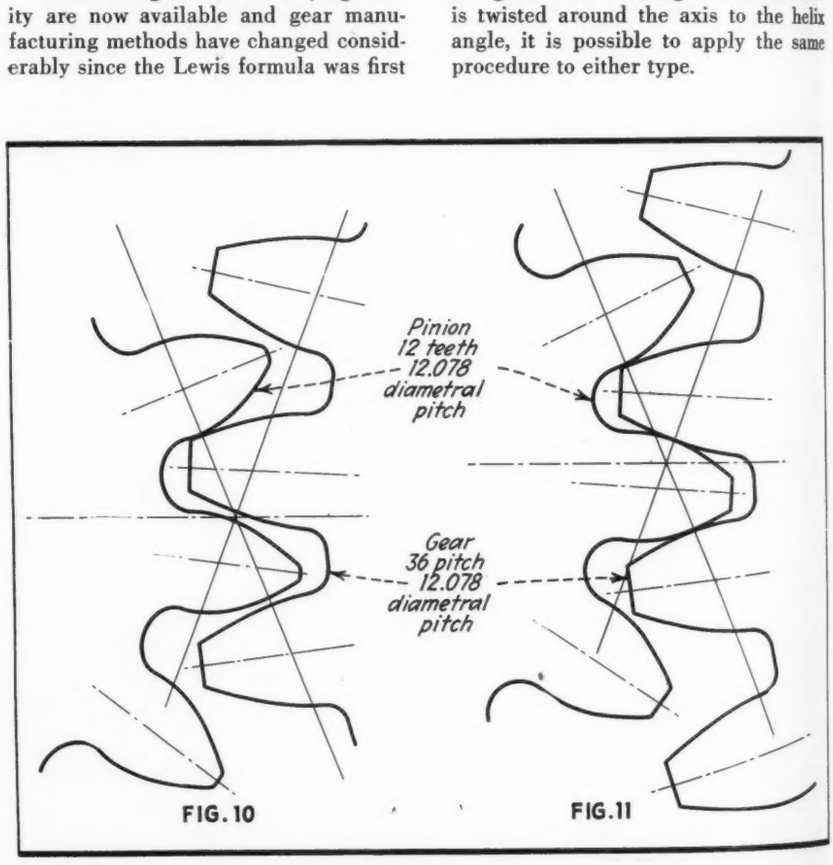


Fig. 10—Strut action exists when a small pinion rolls improperly so that, past the center, only the tip and opposing dedendum touch. Fig. 11—Strut action is corrected by making the pinion teeth oversize so that contact continues along the line of action

WHAT TYPE OF PLASTICS?

Compression vs. Injection Molding

D. A. DEARLE

KALEIDOSCOPIC progress of the plastics industry and the development of new molding materials during the past few years apparently has created confusion in the minds of many product engineers as to the relative advantages of thermosetting and thermoplastic molding compounds. Plastics molders continually are bombarded with questions asked by engineers who are acquainted only with fundamentals of materials and molding technique. Some of these questions which appear time and time again are collected and answered in this article.

QUESTION 1: Much is heard about thermoplastic and thermosetting materials. Just what is the difference between the two compounds?

Ever since the introduction of the injection process of molding, the thermoplastic materials have gained more prominence. These plastics are known as Lumarith, Plastacele, Tenite, Crystalite, Plexiglas and other trade names, and include cellulose acetate, polystyrene, methyl methacrylate among the most widely used compounds. Articles molded from these raw materials can be made in an unlimited number of pastel shades as well as transparent colors and water clear. In molding, thermoplastic materials are heated to a plastic state and then forced under high pressures through a constricted nozzle into a relatively cold mold where solidification into the desired shape takes place. The materials undergo only a physical change during the process.

Thermosetting compounds are molded quite differently. The powder is usually loaded manually into a heated die; this mold is subsequently closed and high pressure is applied by means of a compression press. With the combined application of heat and pressure the material is then cured for the required length of time and finally ejected by either mechanical means or compressed air. The fundamental difference between thermosetting compounds and the thermoplastic group lies in the fact that the former undergo a definite chemical change in the process of molding. Thermosetting materials are usually phenol-formaldehydes or urea-formaldehydes and are known by many trade names some of which are Resinox, Bakelite, Plaskon, Beetle and Durez. Articles made from these materials withstand greater heat, are usually more brittle, cost less per pound, but take longer to cure. They are not available in water-clear colors, although the so-called cast phenolics may be obtained to a limited extent in amber or some other dark transparent color.

QUESTION 2: Why are tool costs so much less for injection molds than dies made for compression molding?

On a per cavity basis, tool costs are not less for injection molds. It is only because of the rapidity with which injection molding can be conducted that this erroneous impression has been created. In other words, by use of an injection mold the rate of production can often be increased three or four fold, and consequently the number of die cavities for a given production rate can be reduced proportionately. Savings are often realized in the original mold investment when fewer die cavities can be utilized. That is the reason for the belief that injection molds cost less than compression dies. Actually the same type of steel goes into an injection die as that used for compression molds, and the workmanship in either type is of the same quality. As a matter of fact, the knockout mechanism in many injection molds is far more complicated than the ordinary means of ejection found on compression dies. As a general rule tool costs are no less for injection molds on the basis of price per cavity.

QUESTION 3. In compression molding, certain designated cavities can be blocked off by the simple method of not loading them with powder. Is it possible to accomplish this in injection molding?

If it is desired to block off one or more cavities on an injection mold it can be done—perhaps not quite as simply as in conventional molding.

In compression molding the control of powder supply is entirely in the hands of the operator and he can do whatever he pleases with regard to the amount of compound which is placed in each cavity, because the material is added first and the mold subsequently closed. Injection molding operations, however, are just the reverse; in these the die closes and then the material is forced into the cavities.

Injection dies of the direct runner type are all constructed with channels through which the liquid material has to flow before reaching the cavity. This flow emanates from the heating cylinder, passes through the sprue hole whence it goes into the runners, then through the gates into the piece. Inasmuch as the individual cavities are made with a leader which is lined up to meet the main runners, the flow of material can be stopped by merely plugging this or that particular leader. The same result can be accomplished by turning or reversing a round cavity so that the leader does not coincide with the main channeling. Such a change, of course, involves removal of the mold from the press, a procedure unnecessary in blocking off cavities in compression molding.

QUESTION 4: Is it possible to obtain parts of different design from the same die in injection molding?

There are innumerable instances where a product requires more than one plastic molded part to make a complete unit. This condition may be found in the form of a right and left half, or perhaps a top and bottom. In order to economize in tool costs, a combination mold becomes advisable. Injection molds can be designed with one or more parts in the same die, but it is preferable to balance the volume of flow as much as possible so as to distribute the material uniformly. The gating of the die must be proportionate to the size of the piece in each instance. Excellent examples of multiple designs in the same mold are to be found in the manufacture of plastic novelty parts, hacksaw handles, toy whistles and other products.

QUESTION 5: Assuming that an article is found to be practical from a mold design standpoint, can this item be made in any color such as black for one run, then transparent for another?

If the die for producing the parts is made for injection molding, the color possibilities are unlimited. Changing

from one color to another involves the loss of a certain amount of material, but any shade can be made from the water clear to the jet black. The product, of course, has to be made from one or another of the thermoplastics.

Color possibilities in compression molded parts are restricted somewhat by the materials available. A full range of pastel shades can be obtained, but the water clear or transparents can not be made under normal circumstances.

QUESTION 6: If the owner of a compression mold wishes to have his product made by the injection method, can the die be used in either type of press?

Compression and injection molds are not interchangeable from one type of machine to the other, and therefore it can not be assumed that such dies can alternate. In many instances a compression mold can be converted into the injection type by utilizing the numerous cavities and setting them into a new frame. The amount of work and cost involved depends on the number of cavities, their design, the knockout mechanism and other factors. When such a change is contemplated it is always worthwhile to investigate the cost compared with the price of an entirely new mold.

QUESTION 7: Why are large parts made of thermoplastic materials so much more expensive that the same ones made from thermosetting compounds:

All costs, whether in the plastics or any other industry, are based on three main factors: material, labor, and overhead. Thermoplastic raw materials are almost four times more expensive than the cheapest thermosetting powders. This fact in itself is the chief reason for the size limitations of the acetates, polystyrenes, and acrylates. Until a reduction in the price of thermoplastic raw materials become effective the large parts will continue to cost considerably more than the same items in phenolic.

While the thermoplastic material price is much higher, the cost of the injection molding operation is lower because of the greater speed of the press cycle. Compression molding cycles are seldom less than one minute and usually average two or three minutes to the heat. With injection molding the cycle may be as low as three shots to the minute and seldom exceeds one minute. To offset this advantage, however, the molder usually operates fewer cavities in order to economize on mold expenditures. Hence, the saving in labor fails to absorb the exorbitant excess of raw material cost.

QUESTION 8: In the event that it is desired to increase production on either an injection or compression mold, can this be accomplished by merely adding extra cavities, or must another die be constructed?

The size and design of the injection mold determines to a large extent the possibility of adding more cavities. If the die is fairly small and permits the extension of the main runner, then it is reasonable to assume that the capacity can be increased. It should be kept in mind that a new frame has to be constructed and that the old cavities have to be utilized along with the new ones in the formation of a larger die. The limits of this expansion are governed by the size of the injection machine platen. If it were found desirable to increase the production capacity by the addition of a few more cavities it would undoubtedly prove more economical to make an entirely new mold and salvage the cavities from the old frame. Where hobs were used, these hobs could form the additional cavities.

On a compression mold it is feasible to add more cavities to a die, but the construction of a new frame usually becomes necessary. In increasing either type of mold, additional knockout bars and pins have to be made to take care of the ejection of the additional parts. Assuming that the cavities are in good shape and the press has sufficient capacity to adequately supply the required pressure, there should be no need for building a new mold to increase production.

QUESTION 9: There are many instances where plastics could be used advantageously except for the fact that the parts would not have sufficient mechanical strength to withstand various physical tests. Can this deficiency be overcome by reinforcing plastics with metal in either the injection or compression method of molding?

Again, the specific design of the item has to be taken into consideration. There are innumerable instances where plastic parts have been strengthened by the use of metal inserts, but on the other hand, the shape of a piece often precludes any possibility of reinforcement by the use of metal. Take for example an ordinary injection molded button. If it were desired to strengthen such a part by the use of a metal disk, this could only be accomplished if the center of the button were permitted to show the metal. With such a design the metal insert could be mounted on a center stud and the material forced around it after the dies were closed. To completely conceal the metal would be impossible inasmuch as it would necessitate a process of "floating" the insert.

Where a certain portion of the metal can extend beyond the limits of the mold cavity itself, an anchorage is made possible and reinforcement can be successfully accomplished. Plastics have been molded over metal parts such as handles on the shanks of knives, insulation on switch toggles, coverings around threaded inserts and other parts, but the complete coverage of a piece of metal is highly impractical. This, of course, does not refer to the spraying of metal parts which is a different process altogether.

The same principles hold true with compression molding except perhaps that it becomes more difficult to hold metal inserts in accurate positions when utilizing the thermosetting materials. This is because the flow of material commences prior to the complete closure of the die.

QUESTION 10: Which type of molding affords the greater opportunity for producing parts that have side holes, exterior undercuts or diagonal holes?

In general, it is easier to mold parts with side holes and irregular contours by injection molding, because generally there are fewer cavities in an injection mold with only two rows of impressions. With this type of frame construction it becomes possible to apply mechanical means on either side of the die for the removal of side hole mold pins and die parts for forming exterior undercuts. This means that a product can be designed to include certain forms of undercut and the mold can be so constructed as to automatically relieve the part while the machine is in operation.

The average compression mold has more cavities in order to achieve high production at much slower cycle time. Hence the use of automatic side hole relief for each cavity is usually too expensive.

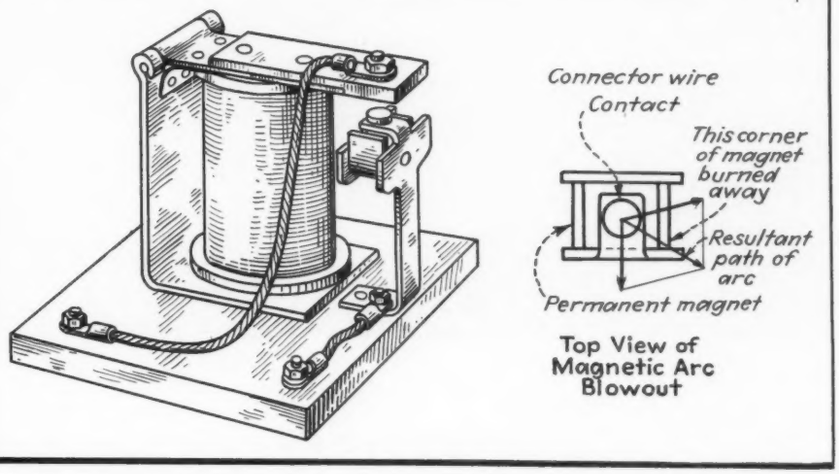
QUESTION 11: Often it is desired to cement two plastic molded pieces together. Are both the thermosetting and thermoplastic products tractable to this process?

Thermosetting plastics are not affected by organic solvents and hence have to be cemented by means of commercial adhesives such as glue. This type of joint, while generally satisfactory, is not of a permanent nature and can not be relied upon to last indefinitely.

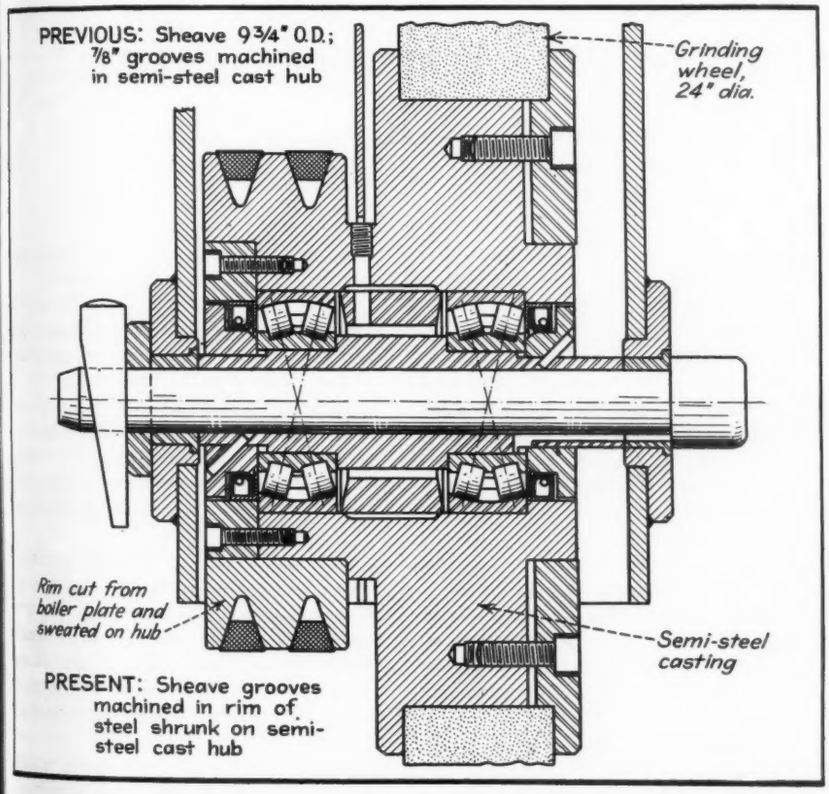
Thermoplastic materials, on the other hand, are soluble in ketones and acetones. Hence a strong and permanent bond can be obtained by the use of their respective solvents. As a result of this bonding property of acetates and other thermoplastics it becomes possible to design many injection molded parts in two halves, often gaining the benefit of reduced weight.

Causes and Cures

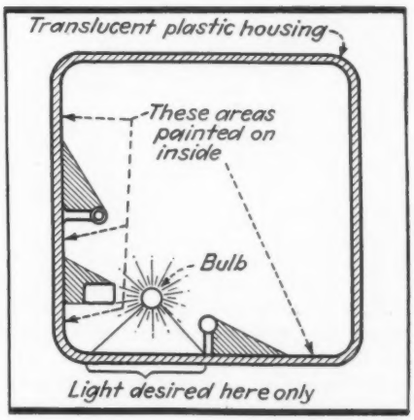
Magnetic arc blowout on a small relay began to burn off one corner of the permanent magnet around the contacts. Investigation disclosed that the spark, instead of blowing straight out, was blowing out at a tangent. Cause: connector wire from the upper contact to the terminal was so close to contacts that magnetic field set up by current in wire also acted upon spark. Vector sum of two magnetic fields, with high current flowing, blew spark out across the corner of the magnet. Cure was to change position of the connector wire so that it led back over the hinge. Designs of all electrical equipment should be checked to eliminate undesirable magnetic effects of current-carrying wires and other stray magnetic fields.



Top View of Magnetic Arc Blowout

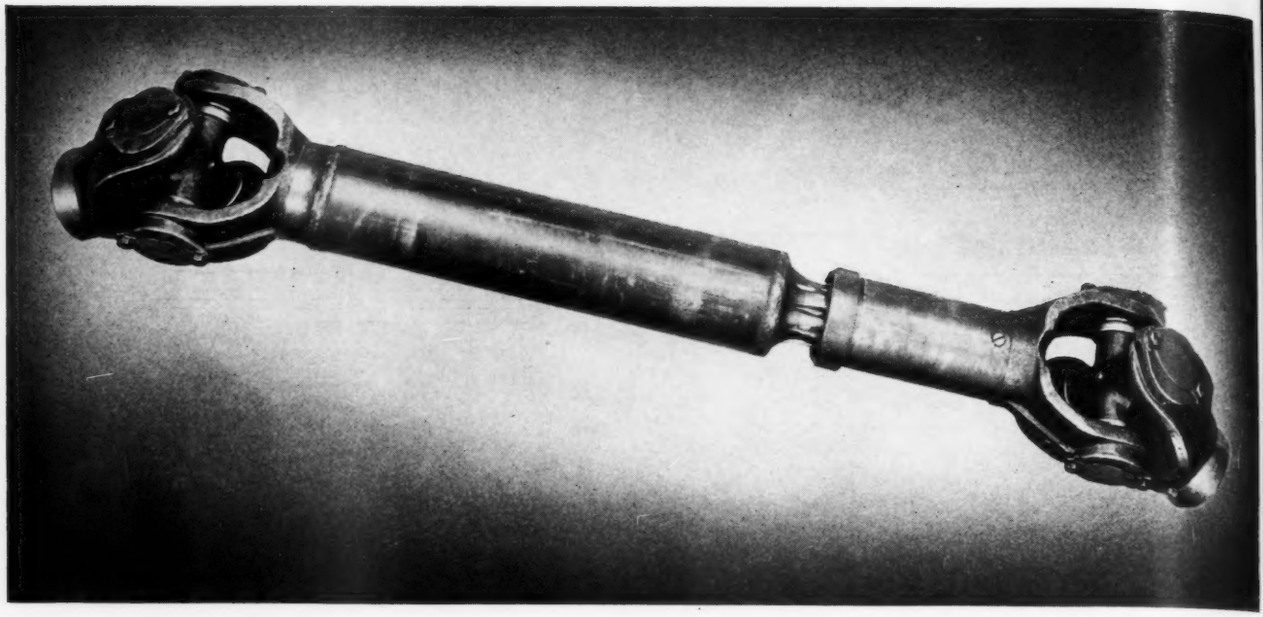

PREVIOUS: Sheave 9 3/4" O.D.; 7/8" grooves machined in semi-steel cast hub
Grinding wheel, 24" dia.
Rim cut from boiler plate and sweated on hub
Semi-steel casting
PRESENT: Sheave grooves machined in rim of steel shrunk on semi-steel cast hub

Blowholes in semi-steel cast hub for 24-in. dia. grinding wheel caused trouble in the grooves for V-belt pulley. Blowholes always turned up when machining the grooves, and their sharp edges shortened the life of the belt. Cure was to make the hub in two pieces, as shown in the lower half of the drawing. Pulley rim is now torch-cut from $2\frac{13}{16}$ in. steel plate, machined on inside diameter to 6.250/6.251 in. and assembled to semi-steel cast hub by shrink fit. Hub is machined to diameter of 6.257/6.256 in. Pulley grooves cut in the steel rim are smooth and prolong belt life.



Shadows cast by dial light and tube filaments on inside of translucent urea-formaldehyde plastic radio case showed through, creating unattractive light and dark patches on the light-colored case. Urea-formaldehyde had been chosen for its attractive depth of color. This problem of shadows has arisen in many electrical appliances and accessories where lights are mounted inside plastic housings. Appearance of the radio case was greatly improved by spraying the inside of the housing with an opaque paint in places where undesirable light showed through. Another effective cure is to place a small metal shield around each bulb to confine the light to areas where it is desired.

PRODUCT ENGINEERING will pay a minimum of $3 for each example published in Causes and Cures. Where illustrations are necessary, include drawings, rough sketches or photographs.



UNIVERSAL JOINTS

How to Determine Loads by Vectorial Analysis

L. D. HAGENBOOK, *Assistant Chief Engineer*
J. H. HOLSTEIN, *Designing Engineer*
Goodman Manufacturing Company

TRANSMISSION of power through the Hooke's universal coupling produces a number of important secondary effects which cannot be neglected if a successful design is to be made. Calculation of torques and bearing loads for any universal joint application must consider that the angles between the shafts and the cross-arm member continually change, hence the loads vary from minimum to maximum values as the shafts rotate. Also, the varying speed of the driven shaft creates important inertia effects that should be considered.

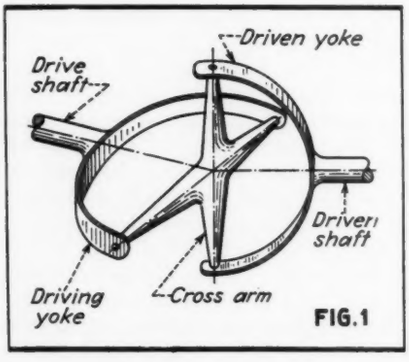


Fig. 1—Simplified sketch of Hooke's universal joint for coupling shafts whose axes are not aligned

This article presents results of a study made to obtain design data, and shows how to use the mathematical shortcuts of vectorial analysis in solving universal joints. Advantage of torque vector analysis is that it circumvents the laborious summation of moments and forces otherwise necessary to obtain equilibrium at every point. Loads on any one bearing can be solved independently of any other bearing, eliminating danger of cumulative error. It is also possible to check any formulas or figures by reversing the procedure.

Calculation of Torques

The cross arm of the universal joint shown in Fig. 1 is free to rotate about two perpendicular axes and therefore can transmit torque only on the axis perpendicular to the two arms. This fact is the starting point for analysis; the cross arm member will be taken as the free body. It is in equilibrium by the action of two equal and opposite couples or torques whose axes are normal to the plane of the arms.

Torque transmitted from the cross arms to each of the shafts can be separated vectorially into two components as shown in Fig. 2. One component is parallel to the shaft; this is the torque in the shaft. The other component is normal to the shaft; this is the bending torque or moment produced in the shaft and is counteracted by the bearings and bearing supports.

In Fig. 2 the driving or constant speed shaft has a torque of value T. This produces a torque in the cross arm equal to T sec γ, where γ is the space angle between the axis of the driver and the axis of the cross arm. Likewise, the cross arm transmits its torque of T sec γ to the driven shaft and produces a torque of T sec γ cos α in that shaft. Angle α is the space angle between the axis of the cross arm and axis of the driven shaft.

Now the angles γ and α are determined by two related conditions: the angle β subtended by the center lines of the two shaft, and the angle θ, angle of rotation of the driving yoke measured from a position in which the axis

of the driving yoke is normal to the plane of the two shafts. Fig. 3 diagrammatically illustrates these angles, and shows the paths of the cross arms. For convenience, angles in the following work are measured in degrees.

By spherical trigonometry it can be shown that:

$$\frac{\sin\gamma}{\cos\phi} = \frac{\sin\alpha}{\sin\theta} = \frac{\sin\beta}{\sin 90^\circ}$$

Also that:

$$\tan\phi = \cos\beta \tan\theta$$

By the use of these equations the values of torques represented in Fig. 2 can be solved in terms of β and θ, and the following important equations can be obtained:

Bending moment on driver:

$$T\tan\gamma = T\tan\beta\cos\theta \qquad (1)$$

Torque in cross arm:

$$T\sec\gamma = T\sqrt{1+\tan^2\beta\cos^2\theta} \qquad (2)$$

Bending moment on driven shaft:

$$T\sec\gamma\sin\alpha = T\sin\beta\sin\theta\sqrt{1+\tan^2\beta\cos^2\theta} \qquad (3)$$

Torque in driven shaft:

$$T\sec\gamma\cos\alpha = T\left(\frac{1-\sin^2\beta\sin^2\theta}{\cos\beta}\right) \qquad (4)$$

The maximum values of the above four equations are:

Maximum bending moment on driver:

$$T\tan\gamma = T\tan\beta, \text{ when } \theta = 0^\circ \qquad (1a)$$

Maximum torque in cross arm:

$$T\sec\gamma = T\sec\beta, \text{ when } \theta = 0^\circ \qquad (2a)$$

Maximum bending moment on driven shaft:

$$T\sec\gamma\sin\alpha = T\sin\beta, \text{ when } \theta = 90^\circ \qquad (3a)$$

Maximum torque in driven shaft:

$$T\sec\gamma\cos\alpha = T/\cos\beta, \text{ when } \theta = 0^\circ \qquad (4a)$$

The bending moments $T\tan\gamma$ and $T\sec\gamma\sin\alpha$ are not absorbed by the torsional resistance of the shafts and are, therefore, the moments which determine the radial load upon the external bearing supports.

Torque Resulting from Inertia

When a single Hooke's universal joint revolves, the driven member has a variable angular velocity. An equation for this can be derived from Equation (4) which expresses the relation between the torques of the driver and driven shaft. Since power in both shafts is equal, the angular velocity of the driven shaft is:

$$\frac{\cos\beta}{1-\sin^2\beta\sin^2\theta}\times\omega$$

where ω is the angular velocity of the driver. The variation in velocity produces an angular acceleration, an equation for which is found by taking the derivative:

$$\frac{2\cos\beta\sin^2\beta\cos\theta\sin\theta}{(1-\sin^2\beta\sin^2\theta)^2}\times\omega^2$$

This acceleration reaches a maximum at:

$$\sin^2\theta = \frac{1}{4\sin^2\beta}\left[3\sin^2\beta - 2 \pm \sqrt{(3\sin^2\beta-2)^2+8\sin^2\beta}\right]$$

determined by again taking the derivative and equating to zero.

For all practical purposes the acceleration reaches a maximum at $\theta = 45^\circ$, and has a value of:

$$\frac{\cos\beta\sin^2\beta}{\left(1-\frac{\sin^2\beta}{2}\right)^2}\times\omega^2$$

The maximum angular acceleration multiplied by the Wr^2/g of the driven member gives the maximum value of the pulsating torque created by the inertia of the driven member. If the angular velocity ω is expressed in radians per second, the radius r in feet and W in pounds, then the torque will be in pound-feet.

If inertia torque should exceed the torque of the driver it will create a reversal of the load and cause a rattle in any loose joints. It is for this reason that minimum clearances should be specified for all joints of a mechanism incorporating a high speed universal joint. Furthermore, this pulsating torque demanded by the variable speed element will cause some slight variation in the so-called constant speed element, the magnitude of which variation is dependent on the relative polar moments of inertia of the elements.

Loads in Yoke Bearings

Usually two couplings are employed in a drive and are placed so that the final shaft has constant speed. This requires (1) that all three shafts lie in the same plane, (2) that the angle between the input shaft and the intermediate shaft be equal to the angle between the intermediate shaft and the output shaft, and (3) that the yokes on the intermediate shaft lie in the

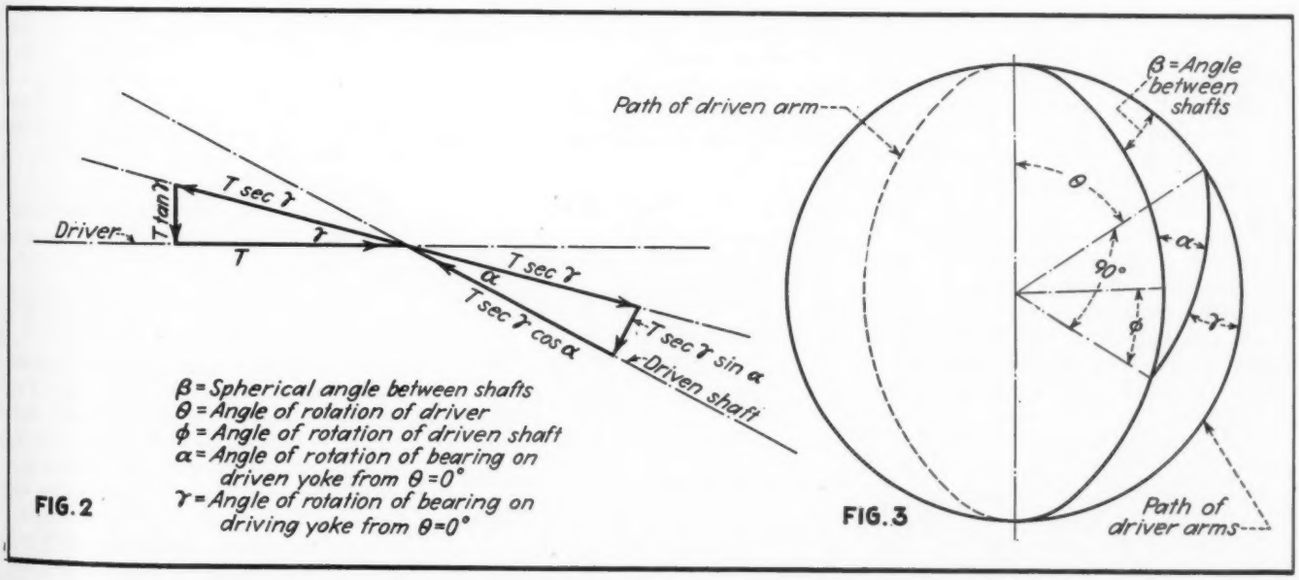


Fig. 2—Torques transmitted from one shaft to the other through the cross arm are split into two components: torque in the shaft and bending moment acting perpendicular to shaft. Bending moments in each shaft are in the planes of respective yokes, so torque vectors are drawn perpendicular to yokes. Fig. 3—Diagram of paths of yoke arms for driver and driven shafts illustrates changing angles which cause varying loads and varying speeds in the cross arm, shafts and bearings

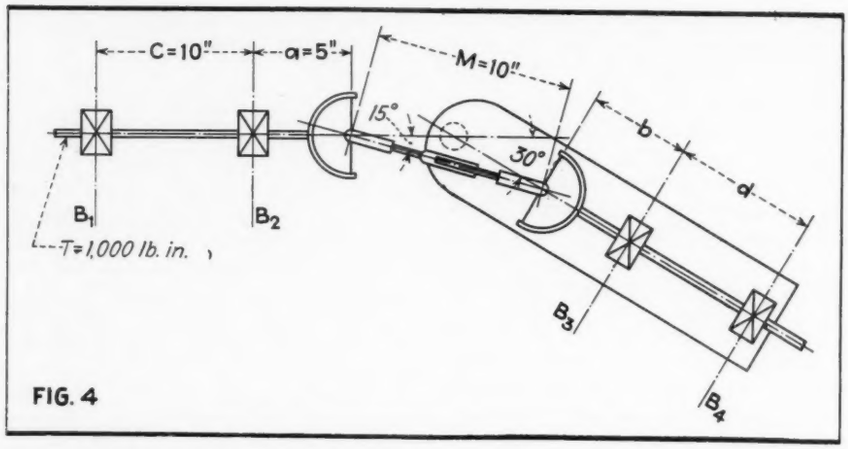


FIG. 4

Fig. 4—Typical application of universal joints uses two couplings to give uniform speed of driven shaft. Example worked out in text shows how to calculate bearing loads

same plane. Fig. 4 illustrates an arrangement satisfying these conditions.

The above requirements are also met when the input and output shafts are parallel but offset. If the shafts are parallel the bending moments acting on the middle section are equal and opposite. However, if the shafts are not parallel there will be an unbalanced remainder which has to be supported by thrust bearings on the ends of the cross arms. The thrust load on any one bearing of the cross arm in turn is carried as radial loads through the two adjacent bearings on the cross arm.

In the case of the drive shown in Fig. 4 the bending moments are equal and additive and are balanced by thrust loads on the yokes of the middle section. From Equation (3a), this thrust has a maximum value of:

$$\frac{2T \sin \beta}{M}$$

where M equals the length of the intermediate shaft.

This formula shows the desirability of keeping the middle section M as long as possible when angle β has any appreciable value. Also, since the thrust in any one yoke bearing reverses every 180 deg., the clearance on the thrust bearings should be made slightly negative to allow for spring in the yoke.

The resulting radial load on the yoke bearings for such a drive as shown in Fig. 4 is found from Equations (2) and (3) as:

$$T\left[\frac{\sqrt{1+\tan^2\beta\cos^2\theta}}{D} \pm \frac{\sin\beta\sin\theta}{M}\sqrt{1+\tan^2\beta\cos^2\theta}\right]$$

where D is the distance between the bearings of a yoke. The maximum value of this equation is best found by plotting a curve.

Loads in Shaft Bearings

It is not within the scope of this article to set forth the formulas for all applications of universal joints. However, the following example will illustrate the general procedure in applying the formulas vectorially derived to the determination of loads in the shaft bearings.

In the application illustrated in Fig. 4, the actual load on bearing B_2 is imposed by a bending moment in the driving shaft and by a bending moment in the intermediate shaft. The bending moment in the drive shaft is resisted by a couple whose arm is c acting through the bearings. Bending moment in the intermediate shaft is the sum of the equal bending moments set up by the joints at both ends. This moment produces a radial load on the bearings of the driver yoke which in turn produces loads on the external bearings. Bearing loads set up by these bending moments, from Equations (1) and (3), are added by quadrature to give actual load on bearing B_2:

$$\text{Load } B_2 = \frac{T}{c}\sqrt{\tan^2\beta\cos^2\theta + 4\sin^2\beta\sin^2\theta\,(1+\tan^2\beta\cos^2\theta)\left(\frac{c+a}{M}\right)^2}$$

The maximum value for which is found by calculus to occur at:

$$\sin\theta = \frac{1}{\sin\beta}\sqrt{\frac{4\left(\frac{c+a}{M}\right)^2 - 1}{8\left(\frac{c+a}{M}\right)^2}}$$

substituting the values from Fig. 4:

$$\sin\theta = \frac{1}{0.25882}\sqrt{\frac{4\times 2.25 - 1}{8\times 2.25}}$$

$$= \frac{1}{0.25882}\times\frac{2}{3} = 2.5765$$

Obviously there is no angle whose sine is greater than unity, so the equation has no solution. This means that at no time is the combined load from the two sources greater than the maximum of one or the other of them alone. In the above calculation the fact that the load from the intermediate shaft is not always normal to the driver has been neglected. However, the result is the same in this case.

The maximum load from the bending moment in the driver, from Equation (1a) is:

$$\frac{T\tan\beta}{c} = \frac{1{,}000\times 0.268}{10} = 26.8 \text{ lb.}$$

and the maximum load from the bending moment in the intermediate shaft, from Equation (3a) is

$$\frac{2\,T\sin\beta}{M}\left(\frac{c+a}{c}\right)$$

$$= \frac{2\times 1{,}000\times 0.259}{10}\left(\frac{10+5}{10}\right) = 77.7 \text{ lb.}$$

Therefore, the maximum load on bearing B_2 is 77.7 lb. In this example the intermediate shaft has a sliding spline connection, which will cause a pulsating thrust load on either B_1 or B_3 depending upon which is held.

Horizontal and Vertical Loads

In certain types of supports the horizontal and vertical components of the bending moments are carried by different bearings. When calculating these bearing loads, it is convenient to remember that the bending moment imposed upon the driving shaft is in the plane of the driving yoke and similarly the bending moment on the driven shaft is in the plane of the driven yoke. The torque vectors, of course, are normal to the yokes.

Assuming as in Fig. 3 that both the driving and driven shafts lie in the horizontal plane, the horizontal and vertical components of the bending torque vectors are determined from Equations (1) and (3) as follows:

For driving shaft:

Vertical component = $T\tan\beta\cos\theta\sin\theta$
Horizontal component = $T\tan\beta\cos^2\theta$

For driven shaft:

Vertical component = $T\tan\beta\cos\theta\sin\theta$
Horizontal component = $T\sin\beta\sin^2\theta$

The algebraic signs have been omitted although it can be seen that the vertical component of $T\tan\gamma$ and the vertical component of $T\sec\gamma\sin\alpha$ are equal and opposite. It should be noted that the vertical components of bending torque or moment produce horizontal bearing loads, and the horizontal components produce vertical bearing loads. The total bearing loads are then found by combining the sum of the horizontal with the sum of the vertical loads by quadrature.

ELECTRIC STRAIN GAGES

For Measuring Loads on Machine Parts

FRANK MITTELBERGER
Mechanical Engineer, R. Hoe & Company

IT IS of basic importance, if economy is to be obtained, to design machinery and structures so that the full strength of the engineering materials is utilized within the margins of safety in order that waste resulting from inefficient design be reduced to a minimum. Many machines and structures are constructed too heavy, in some there is a waste of considerable amounts of steel, copper, nickel and other essential materials. An exact determination of the forces to which machinery is subjected, together with a rational analysis of the stresses and strains caused by these forces, will lead to lighter design which not only will save materials but will reduce the cost of the product and may improve its performance. Such a rational consideration of actual service conditions will put strength where it is needed and will eliminate material where it does not serve a useful purpose.

In a few industries attempts have been made to determine by tests the service loads to which machine parts and structural members are subjected. But in many engineering departments, the forces under which mechanisms and structures have to operate are arrived at by rule of thumb. Stress calculations based on such guess work are of little value and the designer must rely primarily on his experience and his sense of proportions. It is obvious that such a procedure does not approach the scientific methods prevailing in fields of engineering such as thermodynamics and electrotechnics.

A relatively new and simple method for indicating the strain in engineering materials, has made it possible for the mechanical engineer to measure exactly under almost all conceivable conditions, the loads, stresses and strains in mechanisms and structures. In the electric strain gage the designer has an instrument comparable to the thermometer in power engineering or the voltmeter in electrotechnics. The electric strain gage, in making possible actual stress measurements, opens the path for more rational and efficient design based on a knowledge of actual service loads.

The principle of the strain gage is based on the physical property called strain sensitivity, of certain metallic alloys and carbon compounds to change their electrical resistance when subjected to strain. The ratio of unit resistance change to unit strain for the material is known as the strain-sensitivity factor.

Basic Element

The basic element of an electric strain gage can be produced by cementing a piece of thin isolating paper on a metal part and then cementing a small piece of wire having a diameter of about 0.001 in. of strain-sensitive material to the surface of the paper as shown in Fig. 1. (Various strain sensitive materials are listed in *Technical Notes of the National Advisory Committee for Aeronautics*, No. 744.) Both terminals of the wire are then connected with a Wheatstone bridge assembly. In order to determine the strain in a steel rod, like that shown in Fig. 1, the electrical resistance of the strain-sensitive wire is measured first in the unstrained condition and then in the strained condition of the steel rod. The strain in the rod is obtained by simply dividing the unit resistance change by the strain-sensitivity factor. Since the stress distribution in the rod shown in Fig. 1 is obviously one-dimensional the stress can be computed by multiplying the strain by the modulus of elasticity for steel. The applied load, of course, is obtained by multiplying the computed stress by the cross-sectional area of the rod.

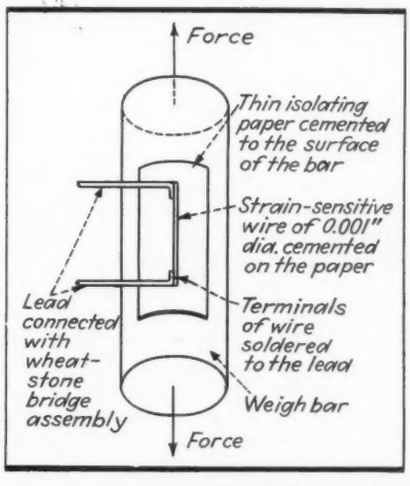

Force
Thin isolating paper cemented to the surface of the bar
Strain-sensitive wire of 0.001" dia. cemented on the paper
Lead connected with wheat-stone bridge assembly
Terminals of wire soldered to the lead
Weigh bar
Force

Fig. 1 — Basic principle of mounting strain-sensitive wire for the measurement of strain

When the stress distribution is two-dimensional the strain in four directions at 45 deg. to each other is determined, and then the magnitude and the direction of the principal stresses are computed. It should be mentioned in this connection that with two-dimensional stress distribution the computation of the stress, when the strains are known, is possible but cannot be carried out by simply multiplying the strain by the modulus of elasticity. An article by H. W. Sibert (*Journal of Aeronautical Sciences*, Vol. 7, November, 1939) shows the method to be employed under these conditions.

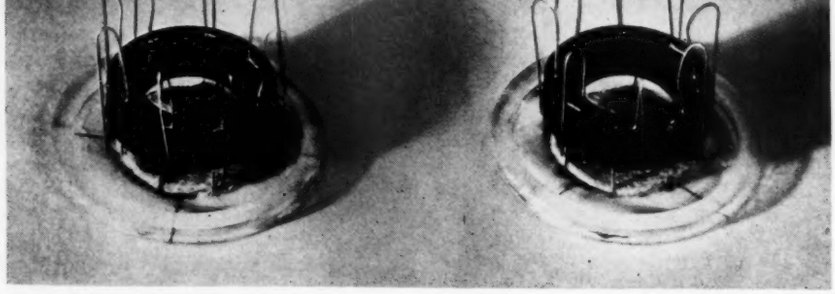
Extensive research work in connection with the development of electric strain gages has been done by Prof. A. V. de-Forest and Prof. A. C. Ruge at the Massachusetts Institute of Technology, as well as by Drs. D. S. Clark, G. Daetwyler and E. E. Simmons at the California Institute of Technology. Among the many difficulties which had to be overcome was the temperature sensitivity of strain-sensitive materials, which means that the resistance of these materials does not only depend on the strain but is also influenced by the temperature. Means for compensating for the effect of temperature changes on strain measurement have been developed.

It was further found that electric strain gages when properly mounted can be used to measure compressive as well as tensile strains, and that the strain-resistance effect has practically no time-lag, which makes this type of gage valuable for the investigation of phenomena of short duration such as vibrations or impact conditions. This is important because mechanical and optical strain gages are always connected with some mechanism, therefore, the readings are often distorted by inertia of the mechanism when indicating a sudden change of strain. The electric strain gage, however, consisting primarily of wire of one-thousandth inch diameter cemented to the surface of the part under investigation has practically no inertia.

An outstanding feature of this gage

Courtesy Baldwin-Southwark

Fig. 2—Metalectric strain gage for measuring strains in one direction. Gage at the left has a removable bar attached to it for protection. After cementing gage in place bar is removed as shown at right



Courtesy Baldwin-Southwark

Fig. 3—Strain rosettes, consisting of four gages to each unit, for determining principal stresses and strains at a given point

is its high sensitivity. Provided with adequate electrical measuring devices it is possible to measure strains as low as one-millionth in. per in., equivalent to a stress of only 30 lb. per sq.in. in steel.

Strain Measurement

Gages for static as well as dynamic strains are available. Most of the static strain gages are temperature-compensated. For dynamic strains, however, a temperature compensation is usually unnecessary since the temperature can be considered constant for the short duration of such strains. Gages measuring strains in only one direction, as well as "strain rosettes," containing 4 gage lines at angles of 45 deg. to each other for two-dimensional testing, are available. In Fig. 2 is shown a strain gage for one-dimensional strain analysis. The steel strip shown on the gage at the left serves as a protecting bracket when the gage is not in use. The terminals, which can be seen on the gage at the right, are connected with the Wheatstone bridge assembly when strains are measured. In Fig. 3 is shown a strain rosette, the eight terminals lead to four gages which are all contained in the unit. They in turn are connected with the electric circuit for strain measurement.

For measuring static strains a Wheatstone bridge assembly as shown in Fig. 4 is used. *G* is a galvanometer of high sensitivity for indicating the balancing of the bridge. Arm *A* represents the strain gage cemented to the surface of the machine part under investigation. Arm *B* is either a high quality precision resistor or a dummy gage of the same strain-sensitive material and having approximately the same resistance as the strain gage in arm *A*. If a dummy gage is used, the strain gage in arm *A* does not necessarily have to be temperature-compensated since the dummy gage, maintaining the same temperature as the strain gage, will compensate for temperature variations. The balancing arms consist of two variable resistors *C* and *D* shunted by a 10,000 ohm fixed and by a 10,000 ohm decade resistance. The shunts can be interchanged by the switch, marked "shunt-reserved." On the panel board of the control box, calibrated strain gages used as variable resistors *C* and *D* are mounted on cantilever beams which are deflected by micrometer screws. The Wheatstone bridge assembly is operated in the conventional way by balancing the bridge for the unstrained and then for the strained condition of the gage. When the resistance change is thus determined, the strain is found by dividing the resistance change by the strain-sensitivity factor. For slowly varying strains an amplifier is substituted for the galvanometer.

A circuit for the measurement of dynamic strains is shown in Fig. 5. Here the voltage drop, caused by the resistance change in the gage, is amplified and then recorded by means of a cathode ray oscillograph. The circuit is calibrated either by closing the switch shunting the resistor r_1, or by the high resistance r_2 shunting the fixed resistance.

Electric strain gages can also be used for the determination of service loads, the qualitative and quantitative determination of vibrations, and for investigations concerning the reactions caused by impact phenomena.

The inertia of mechanical load indicators and the often appreciable displacement necessary to obtain readings make the determination of loads under actual service conditions difficult. By

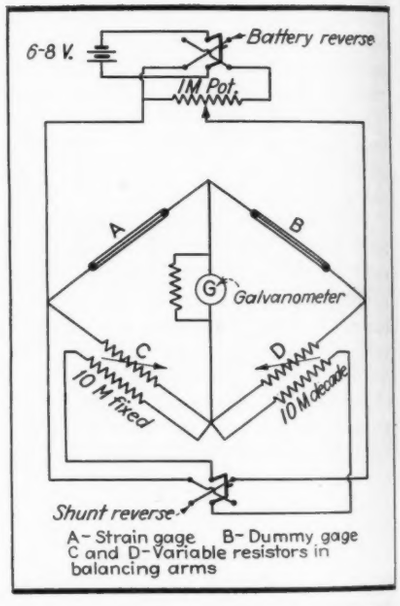


Fig. 4—Circuit of Wheatstone bridge assembly for electric strain gage measurements

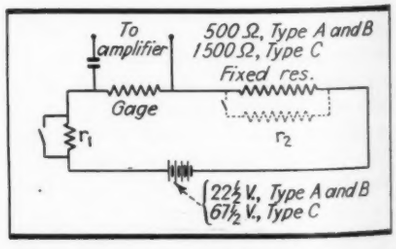


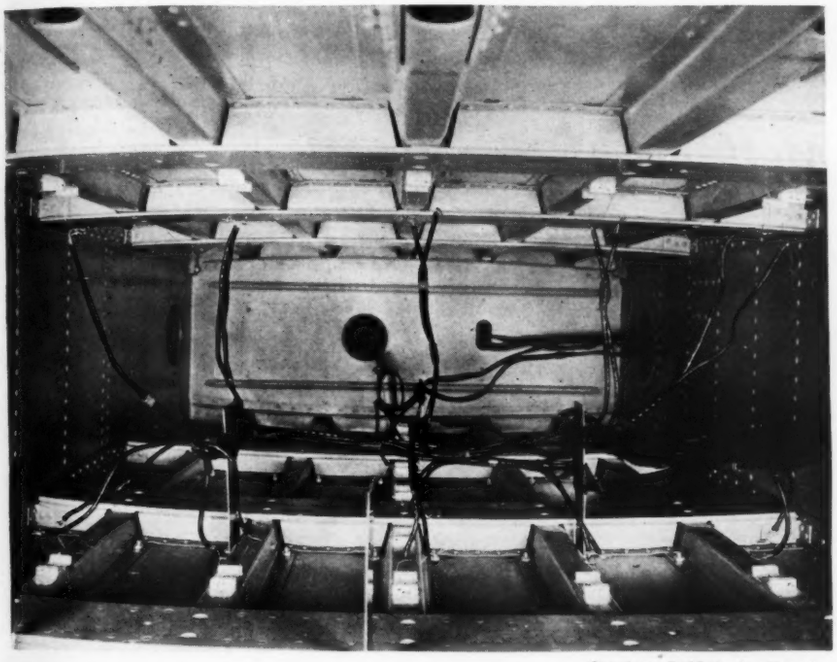
Fig. 5—Circuit for the measurement of dynamic strains

attaching an electric strain gage to a steel rod, a simple weigh bar like that shown in Fig. 1 can be produced which has practically no displacement and the inertia of which is well defined. Such a weigh can, for instance, be attached to the end of an elevator rope for measuring the dynamic loading. Another example for load determination by resistance-type strain gages is their use in connection with the measuring of the forces on cutting tools. Here the gages are directly attached to the tool. After measuring the strains, first the stresses and then the cutting forces are computed, considering the cutting tool as a cantilever subjected to normal loading in combination with a twist movement. Electric strain gages simplify also the determination of forces acting on airplane models in wind tunnels. In this case the gages are attached to the suspension wires of the model. Strain gages are also employed for determining torque in drive shafts.

One method of using strain gages in the aeronautical industry is illustrated in Fig. 6. It indicates how stress analysis by actual strain measurement can be employed to reduce the dead weight of airplanes and thus increase their range of operations. The same method can be used for refining the design of railroad cars, ships, tanks and buildings. This would not only increase the efficiency of the design but would also result in a saving of metals. Wherever mass production of a machine is contemplated, a careful design analysis, aided by actual stress and strain measurement would be worthwhile standard practice.

Electric strain gages having practically no inertia of their own, have been adapted to the measurement of transit strains such as are caused by shock, detonations, or vibrations of high frequency. Fig. 7 is a strain record of a shock wave traveling at the velocity of sound, about 200,000 in. per sec. through a steel bar of stepped diameter. The dimensions of the steel bar and the location of the strain gages are shown in Fig. 8. (This record was obtained with the equipment of the Massachusetts Institute of Technology.) The record indicates the behavior of a shock wave at a discontinuity or shoulder. The record track No. 3 indicates first a large compressive wave passing, and shortly after that a smaller compressive wave. This second wave is the part of the initial wave which was reflected at the shoulder. The third wave shown on record track No. 3 is a small tensile wave which is the reflection of the previous smaller compression wave from the bar's near end. The large tensile wave following next is the returning initial wave which has now propagated through the whole bar to the far end and back and has been re-inforced at the shoulder by the smaller reflection wave mentioned before. The upper record track No. 1 shows first the initial compressive wave passing, then its reflection from the far end as a tensile wave. After about 0.00067 sec. as measured on the time scale there can be seen again a compressive wave passing. This is the reflection of the previous wave coming from the near end and having traveled 134 in. since it has passed last through the bar at the gage No. 1 station.

The record shows that strains lasting for only 1/10,000 seconds are indicated by the electric strain gage. A comparison of a series of quantitative measurements with the theoretical values for the propagation and reflection of impact waves has further proved that the error did not exceed 7 percent, and that most evaluations came as close as 3 percent within the expected theoretical values.



Courtesy Baldwin-Southwark

Fig. 6—Interior of airplane wing showing a number of metalectric strain gages cemented in place for stress analysis work

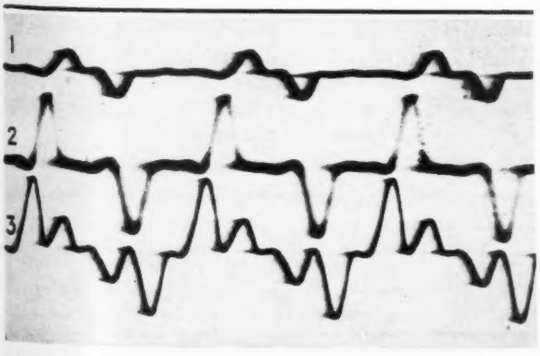


Fig. 7—Strain record of a shock wave passing through a steel bar with a discontinuity

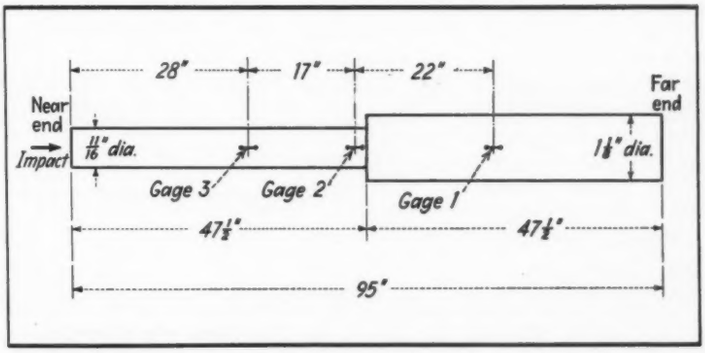


Fig. 8—Steel bar and location of strain gages from which the strain records shown in Fig. 8 were made

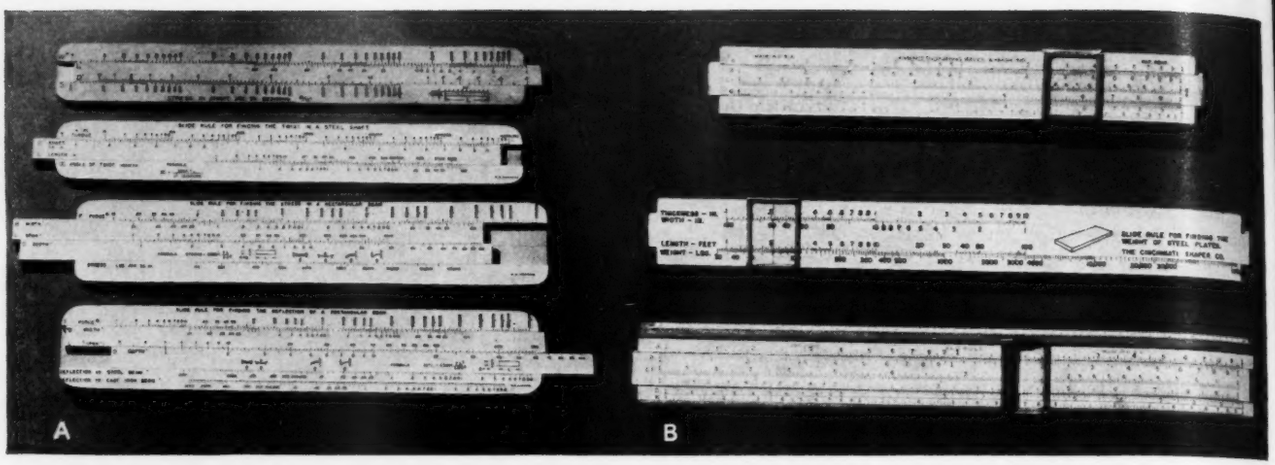


Fig. 1—(A) Four home-made slide rules designed to solve formulas for shaft bending stress and twist, and stress and deflection in rectangular beams. Scales may be applied directly on wood or lithoprinted cardboard scales cemented to slide rule blanks. (B) Inexpensive single slide rules can be used as blanks for special rules by removing the transparent runners and adding new scales

SPECIAL SLIDE RULES

Simple Methods of Construction

RALPH S. DISERENS
The Cincinnati Shaper Company

SPECIAL SLIDE RULES to handle single formulas save sufficient time in engineering computations to be worth serious consideration. Answers are obtained quickly without the possibility of missing the decimal point. Lack of a simple outline of procedure in their construction may be the chief obstacle to their more widespread use since the theory of slide rule operation is generally understood.

Special slide rules have more than one advantage. They serve the same purpose as alignment or nomographic charts but are often preferred to the latter because of their durability, convenience of operation and in some instances, the reduced chance for error. The biggest advantage in the use of a slide rule made especially for a formula, as with nomographs, is that it places the decimal point in the answer, thus eliminating this possibility of error. It also eliminates extra settings for constants because scales are shifted to take care of all constants automatically.

Thus in functions of variables such as $0.3x^2$, only the value of x is set on the special slide rule. While simple logarithmic scales would be most commonly used on these special rules, just as they are on ordinary slide rules, other scales are also possible for functions, such as $\sqrt{\sin x}$, (0.2416 R log $2R$), log tan x, if appearing in an equation to be added to other functions, Fig. 5 illustrates the construction of such scales.

Basic Principles

It is beyond the scope of this article to go into the details of scale construction. However, a few basic facts will be helpful in preparing special scales. The first is that basically all slide rules do only two things: add and subtract selected scale lengths. The ordinary slide rule adds and subtracts logarithms, but by using logarithmic scales the results of multiplication or division are read directly. The special slide rule is made to handle various functions of variables with attention only to the variable itself, by adjustments in size and position of logarithmic and other special scales.

For addition or subtraction, an equal division scale is used. For multiplication or division, a logarithmic scale is used. In a formula involving an exponent, for example $X = V^2Y$, the scale for V must be twice the size of the chosen fundamental scale, that is, the distance from one to ten on the logarithmic scale for V must be twice the distance from one to ten on the X and Y scales. A coefficient in the formula shifts the scale as a whole without otherwise affecting it.

It is sometimes necessary to combine the addition of variables with the multiplication of other variables in a formula. The simplest method is to perform the addition on the side, and apply the result to a logarithmic scale to be multiplied on the slide rule. Slide rules combining these operations have been constructed, however, and occasionally are worth the extra effort.

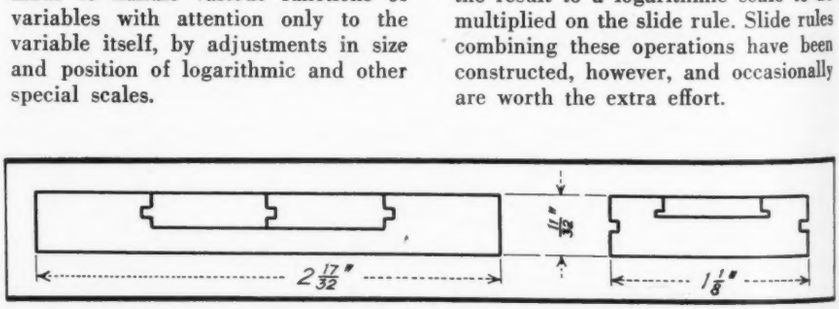


Fig. 2—Shapes used for slide rules. The double slide rule is home made, the single slide rule came from a "Dime" store

One of the "bottlenecks" in the construction of such slide rules — convenient availability of slide rule blanks on which to put the special scales—is now partially relieved by Woolworth's and other chain "Dime and Dollar" stores where single slide rules can be purchased for 25 cents. The faces of these rules, if printed directly on the wood, may be covered over by cardboard scales since the transparent runner is not required on the rules to be described here.

Double slide rules, that is, rules with two sliders, can be made in a woodworking shop. The double slide rules illustrated herewith were home made. The single slide rule blanks can be obtained, if not available at local stores, from two or three manufacturers at low cost or can be "home made" like the double slide rules.

Materials

If the slide rule blanks are to be home made, they are made preferably from two-year, air-dried basswood. Kiln dried wood is less stable than air dried. Maple is used sometimes but it is much more difficult to make the slide fit perfectly. Basswood seems to be affected less by changes of temperature and humidity. Basswood is not as strong as maple, hence a slight twist in a slide does not cause much binding when it is put into the body of the rule.

Both edges of the slide are cut on a high-speed milling machine at one pass. Likewise, the mating edges of the slide rule body are milled with one cut. The slides are milled parallel within 0.001 to 0.005 in. The cutters are set to cut about 0.012 in. smaller than the ways cut in the body of the rule. It was found that after cutting the wood and expecting to get 0.012 in. clearance there was actually only 0.004 in. clearance when the parts were fitted together. This makes a good running fit. The end views in Fig. 2 show two suitable shapes.

The use of cardboard lithoprint scales sprayed with lacquer has proved satisfactory from the standpoint of durability. They have the slick white appearance of the higher-priced rule and are not appreciably affected by temperature and weather conditions. The use of lithoprints has the advantage of making all copies from one original which can be made on tracing cloth with greater ease and accuracy than directly on the wood rule.

Slide rules can also be made with scales inked directly on the wood after treatment to prevent the ink from "bleeding," or sharply defined division lines can be made with a hard pencil. In either case a coat of lacquer will protect the scales.

Construction Procedure

The usual procedure in making the lithoprint type slide rule is as follows:

1. Roughly outline the scales of the rule in pencil on drawing paper in order to be sure that the full range of each variable is covered.

2. Accurately transfer the graduations from logarithmic graph paper to a piece of tracing cloth, using a hard pencil. A number of sizes of logarithmic scales are available on graph paper forms. If a required size is not available it can be produced by projecting a larger scale laid at an angle with the scale line.

3. Ink the cloth tracing.

4. Use the tracing to make lithoprints on ten-ply white cardboard. Fig. 4 shows typical specimens.

5. Cut the cardboard into strips to fit the slides and body of the blank slide rule. A paper cutter insures cutting a straight edge.

6. Glue or cement the cardboard to the slide rule blank, taking care to line up the graduations the same as they appear on the tracing. Shellac makes a good bond. A rubber cement makes the cleanest job because any excess cement can be rubbed off without marring the cardboard.

7. Trim the cardboard flush with outside edges of the rule.

8. Apply lacquer to protect the rule from dirt, preferably by spraying.

9. Apply a little paraffin to the edges of the slide to make it easier.

Scale Arrangement

The formula for calculating the weight of a rectangular steel plate is: Thickness × width × length × constants equal weight, or $TWLK = W$. This becomes in the logarithmic form:

$$\text{Log } T + \log w + \log L + \log k = \log W$$

In Fig. 4 it is shown how these

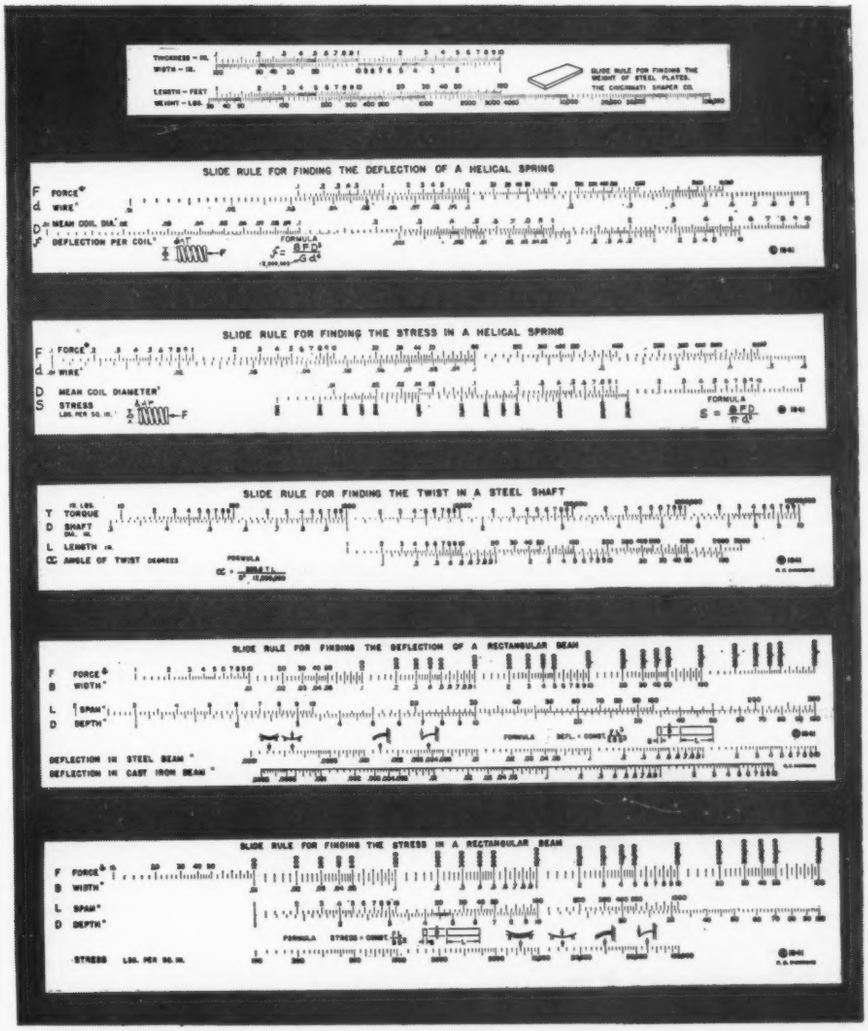

SLIDE RULE FOR FINDING THE DEFLECTION OF A HELICAL SPRING
SLIDE RULE FOR FINDING THE STRESS IN A HELICAL SPRING
SLIDE RULE FOR FINDING THE TWIST IN A STEEL SHAFT
SLIDE RULE FOR FINDING THE DEFLECTION OF A RECTANGULAR BEAM
SLIDE RULE FOR FINDING THE STRESS IN A RECTANGULAR BEAM

Fig. 3—Lithoprints of scales on 10-ply white cardboard for several special slide rules as reproduced from original tracings. When they are cut apart they will be ready to be cemented to slide rule blanks

logarithms are added to obtain the maximum weight that is determinable on the slide rule.

Since all values are raised to the same power, the same size scale can be used for all variables. In this instance a base of 2½ in. for the range from 1 to 10 fitted the rule conveniently and covered all sizes of plates likely to be used. Example 1 in Fig. 3 shows the scales as laid out on white cardboard ready to be cut apart before cementing to the slide rule blank.

The direction of scales on the slide rule is determined by following a simple rule based upon the equation rearranged to equal zero, with the variables in the order that they are to appear on the slide rule from top to bottom.

$$\text{Log } T + \log w + \log L + \log k - \log W = 0$$

Symbols for the variables are then written in this order in the following key:

Key	Scale Direction
← − T + →	T →
← + w − →	← w
← − L + →	L →
← + W − →	W →

Check marks placed opposite the signs of the variables in this form of the equation indicate the directions of all scales. It is usually preferable to so arrange the equation that as many as possible of the scales ascend from left to right, except that the final result should be on the bottom scale of the slide rule.

Deflection of Beam

The formula for computing the deflection of a rectangular steel beam is:

$$\text{Deflection} = \frac{WL^3}{BD^3} \times \text{constant}$$

The equation is written first in the following form with variables in the desired order:

$$\text{Log } W - \log B + 3 \log L - 3 \log D + \log \text{constant} - \log \text{Deflection} = 0$$

The direction of the scales may be determined by the following key. The constant is included when there is an odd number of variables because the variables must be arranged in pairs, two for each slider edge. If no constant were indicated in the above equation the constant 1 would be supplied, or zero where several quantities are added and subtracted. Thus log 1 equals 0 in the logarithmic form of the equation.

Key	Scale Direction
← − W + →	W →
← + B − →	B →
← − L + →	L →
← + D − →	D →
← − K + →	K →
← +Def.− →	Def. →

The check marks, which indicate the directions of scales for each variable, were placed with respect to signs in the logarithmic equation above. In this instance all of the scales ascend from left to right. This is not always practical because variables must also be arranged in a logical order for handling the equation.

The scales for L and D on this slide rule must be three times the size of the basic scale chosen for the others. A base of 6 in. for L and D, and 2 in. for the others was found convenient. These sizes are chosen with respect to range of the variables. Thus, the desired range for L was 2 to 300. If it had been 2 to 3,000 the scale size could have been only about two-thirds as large. The longest scale controls the size of all.

This can be determined by a study of desired ranges of variables, their scale sizes and displacements, if any. The scales for determining deflection of a rectangular beam are shown next to the bottom in Fig. 3 ready to be cut apart and cemented on a slide rule blank. The completed slide rule is shown at the bottom in Fig. 1(A) set for a solution described below.

Using a rectangular beam formula for determining stress in a cantilever beam, let it be assumed that the beam is 2 in. thick, 4 in. deep, and extends 20 in. from a wall with a 3,500-lb. load at its end. The upper slider is positioned so that 2 in. on the B scale (see Fig. 3) is opposite 3,500 lb. force on the F scale. The lower slider is then moved until 4 in. depth on the D scale is opposite 20 in. on the L scale. The arrow corresponding to cantilever loading then indicates the answer, 0.031 in. deflection for a steel beam or 0.075 in for a cast iron beam.

When the same settings are made on the slide rule for computing stress of rectangular beams the arrow corresponding to cantilever loading indicates 13,300 lb. per sq. in. stress.

Both of these double slide rules can handle equations of four kinds of beams because only the constant is different in these equations.

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The Making of Special Slide Rules, by G. W. Greenwood. Mechanical Engineering, Nov. 1925, p. 1002-6; June 1926 p. 593-4.

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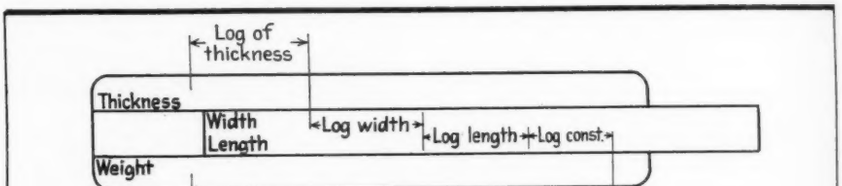


Fig. 4—Principle of addition of logarithms is used in preparing slide rule scales

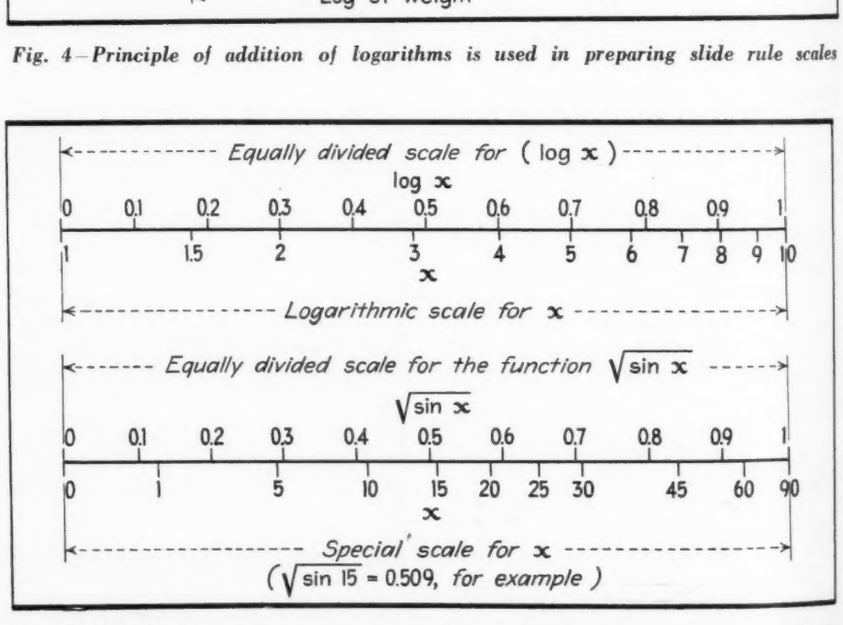

Equally divided scale for (log x)
log x
0 0.1 0.2 0.3 0.4 0.5 0.6 0.7 0.8 0.9 1
1 1.5 2 3 4 5 6 7 8 9 10
x
Logarithmic scale for x
Equally divided scale for the function √sin x
√sin x
0 0.1 0.2 0.3 0.4 0.5 0.6 0.7 0.8 0.9 1
0 1 5 10 15 20 25 30 45 60 90
x
Special scale for x
(√sin 15 = 0.509, for example)

Fig. 5—Method for constructing special scales. The logarithmic scale, shown above for comparison only, is usually taken directly from standard logarithmic graph paper

Materials Section

PEARLITIC MANGANESE STEELS

STEELS WITH LESS MANGANESE are urged by the Office of Production Management because of possible shortages in high-grade imported ores which have been preferred over domestic low-grade ores. This refers particularly to the pearlitic manganese steels containing 0.50 to 2.00 percent manganese, which have grown in use because of their machinability, case hardening properties, strength and toughness, and because manganese is normally one of the cheapest alloying elements.

These steels are not to be confused with the almost unmachinable austenitic manganese steels containing 10 to 16 percent manganese. These important steels use a very small percent of total manganese consumption and no substitute for them has been found.

Substantial reduction in manganese consumption can be made without serious effects on the cost and quality of steel, it has been determined.

Nearly 1,000,000 lb. of ferromanganese now go into steel annually. Manganese content of most steel products has been increasing because consumers have demanded better and better quality of surface, both of steels used in the as-rolled condition and of steels to be converted into finished products.

From 0.15 to 0.30 percent manganese is ordinarily considered essential in steel as a "conditioner" of the molten bath. The commercial "carbon" steels normally carry 0.30 to 0.80 percent manganese without being classed as other than a plain carbon steel. No attention is paid to the effect of the presence of manganese until over 0.50 percent is included. Yet the strength of ordinary structural steel is materially higher because of its manganese content.

GENERAL PROPERTIES

Welding. Pearlitic manganese steels are not "welding steels" but may be welded without difficulty in manufacturing parts involving welding.

Forging. Pearlitic manganese steels present no unusual difficulties in any regular forging operations and they can be freely used in parts requiring forging.

Cold Drawing. Pearlitic manganese steels may be cold drawn without difficulty by standard procedures.

Machining. Pearlitic manganese steels are particularly free machining. Low carbon types are practically equivalent to Bessemer screw stock. Machining becomes more difficult with increase in content of carbon, manganese, and other alloying elements.

Steels containing 2 to 3 percent manganese with other alloys have limited and special uses. They possess unusually good physical properties but there are too many manufacturing difficulties for general adoption.

SUBSTITUTIONS

In many instances merely a reduction in manganese will suffice. Equivalent physical properties may be obtained by substituting other more expensive metals for manganese in many of the higher manganese steels. Alloying, strength-giving effects of manganese may be obtained to some degree by increasing silicon or phosphorous and holding manganese low. Additions of copper and silicon, or copper and molybdenum are effective in cast steels of low manganese content.

An effective, though more expensive substitute for high manganese in wrought steels is molybdenum, or silicon plus molybdenum. Molybdenum is especially effective in steels for heat-treatment by quenching and tempering, because it confers depth-hardening properties. Use of titanium, zirconium, and molybdenum to control sulphur is being investigated.

ENGINEERING STEELS DEPTH-HARDENING

Where manganese is added chiefly to give depth-hardening properties its replacement by silicon is limited by lesser depth-hardening ability. Molybdenum, however, is potent in producing depth-hardening. Roughly speaking, a 0.30 percent manganese, 0.25 percent molybdenum steel in small sections will have most of the characteristics of a 1.30 percent manganese steel upon heat-treatment.

Steels of a wide range of carbon content can be strengthened and given a range of strength and depth-hardening properties suitable for almost any automotive or analogous part, by proper balance of carbon, silicon, molybdenum and of a little secondary aluminum for grain size control, with manganese held at a very low level.

ROLLING AND FORGING PROPERTIES

Manganese content of steels WD X1335, WD X1340, and WD X1345 has two functions. First is to convert most of the sulphur to manganese sulphide and thus improve rolling and forging properties. For that purpose alone the average 1.50 percent content is far too high.

The tendency of high-sulphur steels to break in rolling or forging depends greatly upon carbon content. The tendency is much more apparent in steels containing approximately 0.20

percent carbon than in steels containing 0.40 percent carbon. Steels containing 0.35 to 0.50 percent carbon and 0.075 to 0.15 percent sulphur can be rolled or forged satisfactorily with 1.00 to 1.30 percent, or even 0.80 to 1.10 percent manganese.

HARDENING PROPERTIES

The second function of manganese in the WD X1300 series steels is hardening. Hardness, yield point and tensile strength of such steels are increased in the air-cooled or oil-quenched conditions. It is relatively simple to obtain a yield point of 45,000 lb. per sq.in. in a steel similar to WD X1340 but with only 1.00 to 1.30 percent, or even less manganese.

Use of steels suitable for heat-treatment would save manganese because by heat-treatment, higher physical properties can be obtained with a given manganese content than can be obtained by air cooling.

EFFECT OF SULPHUR

High sulphur content of shell steels has required some manganese to overcome hot-shortness. It is believed that the manganese which combines with sulphur to form manganese sulphide does not contribute to hardness nor yield point. Although sulphur is considered important in determining surface condition and is added to steel to enhance its machinability, the need for such sulphur is open to question. Many machine tools used in manufacture of large-diameter shells will not run fast enough to take full advantage of the extra machinability of high sulphur steels.

COMMERCIAL HIGH-MANGANESE STEELS

Many applications where SAE grades T1330, T1335, T1340, T1345, and T1350 are used would be served equally well by lower than 1.60 to 1.90 percent manganese. Certain specifications, such as those of General Motors Corporation, now call for lower manganese because the carbon content permits rolling reasonably well. Where somewhat greater hardness or hardenability is required, it is believed there is a field for medium carbon sulphurized steels containing 1.44 to 1.30 percent manganese.

Steel Bars and Forging Billets. Most logical substitute for manganese in bars to be heat-treated is carbon with no change in grain size although this might change the heat-treating cycle, lower impact properties somewhat, and decrease machinability in the higher carbon ranges.

Reduction of manganese without changing carbon content would change it from fine to coarse grain steel. There would probably be more surface preparation required and somewhat high rejections for surface defects. In forging the bars there might be increased rejections from cracking in the trimming operations. There would probably be a slight change in properties involving ductility, but somewhat better machinability.

In higher manganese steels containing sulphur it is believed a lower sulphur steel would often perform satisfactorily in machining, except in high-speed automatic machine work.

Sheet Steel and Strip Steel. An average manganese content of 0.35 percent maximum is believed satisfactory.

Wheels, Axles, and Heavy Forgings. No changes in specification requirements for manganese recommended.

Pipe, Lap Weld and Butt Weld. Reduction of 0.25 percent manganese recommended.

Tin Plate. Little can be done to lower manganese content of steel plate to be tin plated because of the possibility of producing bad edges and meeting other difficulties.

Standard Tee Rails. Reduction of manganese is believed practical although with some sacrifice of wearing quality and possibly slight increase in seconds.

CAST STEEL

In alloy cast steel, because of its cheapness, manganese normally runs around 1 percent or over. If other alloys are used they are usually added beside, rather than instead of manganese. Reduction to 0.70 percent is recommended with a few exceptions. Strong cast steels that do not require holding manganese at a high level are well known. Either copper plus silicon or copper plus molybdenum will serve the purpose as well.

Use of phosporus as a strengthening element is not feasible in cast steels. Cast steel does not have the problem of edge cracking on rolling so that copper may be raised to any reasonably desired level. In wrought steel edge cracking occurs with high copper, though it may be avoided by not overheating prior to rolling.

FREE-MACHINING STEELS

An important role of manganese is in the high-sulphur free-machining steels of the SAE 1100, X1100 and X1300 series with sulphur ranging from 0.05 to 0.30 percent with or without phosphorus at 0.09 to 0.13 percent and manganese at 0.60 to 1.65 percent. Manganese is added primarily to prevent cracking during rolling though its strengthening properties and, in steel to be heat-treated, its depth-hardening properties, may be valued also.

Good free-machining properties may be obtained, without appreciable sacrifice of toughness, even in heat-treated alloy steels, by addition of up to 0.25 percent lead, without increasing sulphur and thereby requiring high manganese to balance this sulphur. Machining time on some alloy die steels, produced by a die maker who melts his own steel, is being effectively reduced by addition of lead. Little actual use of this possibility has been made because of complication in warehousing extra grades.

One large user reported that high accuracy screws, pinions, and shafts have been produced on automatic screw machines from leaded LX1112 steel. Its superior machinability reduced the number of screw machines to half what would be necessary with ordinary screw stock. Improvements in steel making may also aid in manganese conservation.

Composition of Commonly Used Pearlitic Manganese Steels

Steel Type	SAE No.	C	Mn	P	S
1	X-1314	0.10–0.20	1.00–1.30	0.045 max.	0.075–0.15
2	X-1315	0.10–0.20	1.30–1.60	0.045 max.	0.075–0.15
3*	.....	0.10–0.20	1.30–1.60	0.04 max.	0.08 max.
4	X-1330	0.25–0.35	1.35–1.65	0.045 max.	0.075–0.15
5	X-1335	0.30–0.40	1.35–1.65	0.045 max.	0.075–0.15
6	X-1340	0.35–0.45	1.35–1.65	0.045 max.	0.075–0.15
7	T-1330	0.25–0.35	1.60–1.90	0.040 max.	0.050 max.
8	T-1335	0.30–0.40	1.60–1.90	0.040 max.	0.050 max.
9	T-1340	0.35–0.45	1.60–1.90	0.040 max.	0.050 max.
10	T-1345	0.40–0.50	1.60–1.90	0.040 max.	0.050 max.
11	T-1350	0.45–0.55	1.60–1.90	0.040 max.	0.050 max.
12*	.....	0.45–0.55	1.00–1.30	0.040 max.	0.08 max.

***Types 3 and 12 are frequently furnished with additions of chromium and/or molybdenum.**

Types 1, 2, and 3 are most commonly used for case hardened parts, such as roller bearings, camshafts, and gears. These steels harden with a hard case at low temperatures and possess a strong, tough core. They all possess superior machining qualities and are classed as free cutting steels.

VANADIUM ALLOYS

Too extensive use of vanadium, which was brought about by substitution of SAE 6100 steels for non-obtainable nickle-containing steels, can be relieved in the majority of applications by using the equally serviceable molybdenum or molybdenum silicon steels with a grain-refining addition of aluminum. This will save chromium as well. Secondary aluminum, or what might be termed "complex ferro-aluminums," made from waste products that cannot be purified economically, may be used for this purpose.

ENGINEERING STEELS

In general, only 0.10 to 0.20 percent vanadium is added to engineering steels and the slight increased cost has been repaid by advantages gained. However, at least some substitution has been made for economic reasons.

STRENGTH

Small amounts of vanadium are used to put steel into a certain condition, rather than for bodily strengthening. The "carbon-vanadium" steels, which have both increased strength and increased toughness over plain carbon steel, normally contain more manganese than plain carbon steel, and it is this that gives the steel its added strength. Vanadium steels are readily cast, forged or rolled.

LOAD-DEFLECTION CHARACTERISTICS

Steel castings containing vanadium have a higher elastic ratio than castings without vanadium but of otherwise like composition and heat-treatment. At the same time, the vanadium castings possess at least equal ductility in tension and considerably higher impact strength and wear resistance. The result is greater ability to sustain stress, or to deform rather than rupture under sudden overloads.

TOUGHNESS AND GRAIN SIZE

Vanadium increases toughness through control of grain size or by producing a fine grained steel. Properties of steels made fine-grained by other means are not wholly identical, but they are acceptable engineering substitutes in many applications. The virtue of vanadium is in its foolproofness; the results are more certain.

In fine-grained engineering steels in small sections, to be quenched and tempered, it may be broadly stated that it is immaterial to the behavior of the finished part whether the fine grain is produced by addition of vanadium, aluminum, zirconium, or titanium plus aluminum, or some combination of these. Total addition of any substitute or combination is only around 0.003 to 0.010 percent.

COMPLEX ADDITION AGENTS

Complex addition agents of nearly every combination of aluminum, titanium, zirconium, and vanadium, as well as silicon have been put on the market as means of obtaining as effective results as with vanadium alone or of reducing the amount of ferro-vanadium required. These complex agents are too new for their action to have been thoroughly appraised. However, some of them, properly used, made a fine-grained, relatively deep-hardening steel instead of the fine-grained shallow-hardening steel produced by single additions handled in the usual way.

Vanadium-free or very low-vanadium steels, containing molybdenum or molybdenum plus silicon, deoxidized with aluminum and given depth-hardening behavior through such additions, have been suggested as substitutes for steels quite heavily alloyed with nickel, chromium, and manganese.

NORMALIZED STEELS

Substitution for vanadium is somewhat more difficult when steel is not to be quenched and tempered, but is to be used in the normalized condition in heavy sections, or to be welded and used without later heat-treatment after it cools from the welding heat.

Heavy shafting and other parts to be normalized are often made of steel with 0.75 to 1.50 percent manganese and 0.35 percent molybdenum; or with 0.75 to 1.50 percent chromium, with normal manganese; or with 1.00 to 1.60 percent manganese plus 0.30 to 0.65 percent chromium, all with about 0.15 to 0.20 percent vanadium. These steels are made up on special order, the tonnage is small and the uses important.

AS-ROLLED AND AS-WELDED PLATE

There are some advantages in using manganese-vanadium steel plate as rolled and for welding. Safe weldability at as high as 0.17 to 0.18 percent carbon, with around 1.20 percent manganese may probably be had in such steel, although the subcommittee on welding low alloy steels of the American Welding Society did not exempt this from its general statement that steel to be welded without preheating or post-heating should be limited to 0.15 percent carbon, whatever its alloy content.

However, a variety of other weldable alloy steels without vanadium are available with adequate high yield strength to be used as substitutes for less exacting conditions.

CAST STEELS

Vanadium steels show high elastic properties and excellent resistance to impact both at ordinary and low temperatures. Use of vanadium steel is advantageous for massive or intricate castings that are not liquid quenched.

At elevated temperatures, vanadium cast steels show considerably better yield strength than carbon steels of like tensile strength. At low temperatures, vanadium cast steels show excellent resistance to impact.

Vanadium is added to various cast alloy steels to prevent formation of too coarse a dendritic structure, one that is difficult to refine in subsequent normalizing. Titanium is as effective as vanadium if suitable ferroalloys of titanium are available. Little hardship would be involved in practically complete avoidance of vanadium in cast steel, with a few special exceptions. Titanium may require a higher normalizing temperature, and according to some authorities, becomes less effective with increase in weight of casting.

PRECAUTIONS

Any substitution of vanadium should be approached with caution and each case studied for its real needs, rather than issuing a blanket edict that no vanadium will be available for wrought engineering steels. In applications where it has been used for convenience and certainty, use of aluminum or titanium and special care in finishing

the steel is recommended. The only practical way to improve the vanadium situation is to work through the makers of alloy steels.

Ultimately, a modification of alloy steel specifications should be brought about to express requirements in terms of grain size and coarsening temperature, hardenability and ultimate mechanical properties rather than to call for "vanadium steel," or some other specified composition in the attempt to secure desired properties.

HOW TO ORDER

If specifications and orders for steel are worded so as merely to require the desired depth hardening properties, grain size, coarsening temperature and mechanical properties without insisting on or indicating the use of vanadium, steel mills can usually meet the requirements without having to use it.

MAJOR USES

Major use of vanadium in wrought engineering steels is in the SAE 6100 series. The lower carbon ranges are used for carburizing, the intermediate ranges for gears, connecting rods, etc., and the higher carbon ranges for springs and certain special parts.

In all these uses other additions to produce fine grain will serve almost as well as vanadium as far as the properties of the finished part are concerned, even if fabrication differs.

MACHINABILITY

It is where intermediate machining must be done that the chief value of vanadium is exerted, in that to coarsen the grain suitably for good machinability, a slight overheating of the vanadium-containing steel will effect the coarsening, whereas with other grain size control agents, more drastic overheating is required. This takes time and fuel and increases danger of warping, scaling, and decarburization. Chromium steels are not as readily decarburized, however, as some other steels, notably silico-manganese spring steel.

Vanadium steels are no more severe upon cutting tools than are other alloy steels of equal strength or hardness. Since the grain of vanadium steels coarsens in a narrow range of temperature, only somewhat above the range employed for normalizing or hardening, machining to a smooth surface is readily attainable by such high temperature normalizing. The fine grain and its associated toughness are recovered again after heat treatment at the usual temperatures.

Machinability requirements can be supplied in fine-grained steel by alloy steel producers without using vanadium if proper coarsening temperatures for the particular substitute are used.

WELDING

Chrome-vanadium steels all require preheating in order to obtain satisfactory results in welding. Sample welds may be required to determine properties obtainable from welding operation and the effect of the heat-treating operation. Gas and arc welding are commonly used, thermit welding occasionally, but resistance welding is not used.

[This material, except under the head of Engineering Steel Uses, is largely abstracted from a recent report by H. W. Gillett for the Metals Conservation and Substitution Group of the Advisory Committee on Metals and Minerals, National Academy of Sciences.]

Properties of Representative Low Alloy Cast Steels with Vanadium

1. 0.22 C, 0.70 Mn, 0.40 Si, 0.20 V ..... (Double normalized and drawn)
2. 0.38 C, 0.87 Mn, 0.32 Si, 0.20 V ..... (Double normalized and drawn)
3. 0.30 C, 0.80 Mn, 0.40 Si, 0.10 V, 1.00 Cr ..... (Double normalized and drawn)
4. 0.42 C, 1.45 Mn, 0.80 Si, 0.38 Cr, 0.16 V ..... (Normalized 1,650 F., drawn 1,150 F.)
5. 0.29 C, 1.02 Mn, 0.31 Si, 1.49 Ni, 0.13 V ..... (Double normalized and drawn)
6. 0.30 C, 1.65 Mn, 0.10 V ..... (Double normalized and drawn)
7. 0.28 C, 1.34 Mn, 0.39 Si, 1.72 Ni, 0.11 V ..... (Annealed)

	1	2	3	4	5	6	7
Tensile strength, lb. per sq. in.	79,000	94,100	94,300	116,000	93,000	101,000	103,000
Yield point, lb. per sq. in.	46,000	64,900	64,850	74,000	65,000	67,000	71,500
Elongation in 2 in., percent	31.0	24.0	27.5	19.0	29.0	27.0	23.0
Reduction of area, percent	58.6	53.9	57.1	43.0	57.8	55.0	40.4
Izod impact value, ft.-lb	50.2	30	.....	.....	55.5	45.0	43.3

ENGINEERING STEEL USES

CARBON VANADIUM STEELS. Principal composition: 0.45-0.55 C, 0.70-0.95 Mn, 0.15 V (min.)

Used for locomotive forgings such as axles, crank pins, connecting rods, and piston rods. Forgings are usually normalized and tempered.

Automobile crankshafts of this steel are drop forged, normalized, restruck and die straightened at normalizing temperature, and then machined without further heat-treatment. These shafts are practically free from internal stresses. This steel responds well to quenching and tempering.

CHROME VANADIUM STEELS. Generally conform to SAE specifications for 6100 series. Steels 6115, 6120, and 6125 are used for case hardened parts, such as automobile gears, camshafts, and piston pins, particularly with carbon range of 0.12-0.20 percent. Steels 6120 and 6125 respond readily to cyanide hardening and are used for gears, bolts, washers, small stampings, and forgings. These steels are also used in boiler construction, superheater tubes, pressure vessels, bolts, tubing for chemical industry, and for welding rods.

Steels 6125 and 6130 are used where high strength and toughness are required, as in automobile axles, steering arms, connecting rods, crankshafts, and for machine tool arbors, spindles, and shafting.

Steels 6135 and 6140 are used in small bevel and spur gears, rocker arms, small tools, wrenches, and other semi-hard parts; also for large forgings where elastic strength combined with toughness and fatigue resistance are required, and internal stresses must be minimized.

Steels 6145 and 6150 are used for highly stressed parts such as transmission gears and for springs.

HYCAR SYNTHETIC RUBBER

Hycar has many of the properties common to the butadiene rubbers, to which class it belongs. As is true for all the butadienes, its properties can be varied by changes in compounding and processing.

OIL AND SOLVENT RESISTANCE. This material is outstanding in its resistance to all products derived from petroleum, vegetable oils, fats, paint and ink driers. Changes in volume on immersion of a specific compound depends upon how it is compounded, the conditions of immersion, and the nature of the oil used. Maximum swell for any all-Hycar stock in any petroleum derivative will probably never exceed 10 percent.

Shrinkage may be obtained with very soft stocks and is caused by removal of the softeners. This can be offset by combining with other materials. Hycar is quite resistant to benzol and carbon tetrachloride, as shown by Table II, but is badly swollen by acetone. It has a high resistance to water as compared with natural rubber and is used where resistance to dilute acids and alkalies is important.

HEAT RESISTANCE. Ordinary compounds have excellent heat resistance. However, special compounds designed specifically for heat resistance are outstanding. Maximum temperatures depend upon conditions of exposure such as whether continuous or intermittent, whether covered with oil or exposed to dry air or steam, and whether flexed, abraded or merely stationary. In a few limited applications the material can be considered for use up to 400 deg. F. Flame resistance is no better than natural rubber.

COLD RESISTANCE. Stiffening occurs at − 45 deg. F. Pliability is recovered immediately when temperature rises.

TENSILE STRENGTH. Ultimate tensile strength in excess of 4,000 lb. per sq. in. can be obtained by proper compounding and vulcanization. Tensile strength tests of typical compounds are shown in Table I. The results are approximately the same as similarly compounded natural rubber stocks, except that pure gum compounds have low tensile strength and are of little commercial use.

ELONGATION. Elongation covers about the same range as that of natural rubber compounds, as Table I shows.

TEAR RESISTANCE. Tear resistance of the best compounds is somewhat inferior to that of the best compounds of natural rubber.

ABRASION RESISTANCE. Under normal conditions abrasion resistance is about 30 percent better than natural rubber. At high temperatures and in the presence of oils, Hycar is considerably superior to natural rubber.

ELASTICITY OR REBOUND. Usually referred to as "snap," this property is generally low. No synthetic equals natural rubber in these qualities.

The capacity to recover from distortion resulting from either tension or compression after an interval of time, however, compares favorably with natural rubber compounds. This property is known as permanent set and is considered of more importance than re-

Table I—Properties of Typical Vulcanized Compounds

Test	Compound Number					
	1	2	3	4	5	6
Shore Durometer hardness						
Original, room temperature	76	59	68	71	30	94
After 48 hours at −25 C. (−23 F.)	96	87	88	96	39	97
After 7 days in oxygen bomb	80	62	75	76	33	95
Elongation, ultimate, percent						
Original, room temperature	485	400	315	705	415	215
After 4 days in oxygen bomb	420	345	345	705	340	220
After 40 hours in air bomb	140	155	125	430	305	...
Modulus at 300 percent elongation, lb. per sq. in.						
Original, room temperature	1,830	1,320	2,040	770	200	...
After 4 days in oxygen bomb	2,220	1,400	2,000	830	210	...
After 40 hours in air bomb	2,300	1,900	.....	2,080	...	...
Tensile strength, ultimate, lb. per sq. in.						
Original, room temperature	3,800	1,840	2,180	3,630	380	2,600
After 4 days in oxygen bomb	3,500	1,790	2,110	3,400	310	2,620
After 40 hours in air bomb	2,300	2,000	2,500	3,210	910	...
Rebound elasticity (Schopper)						
Original, room temperature	13	34	27	13	55	12
After 4 days in oxygen bomb	14	34	27	13	55	12
After 20 hours in air bomb	13	26	22	12	45	12
Compression set						
40 percent constant deflection method, percent	9.31	7.92	8.04	9.9	10.9	11.1
400 lb. per sq. in. constant load, percent	7.84	9.00	9.92	12.1	9.8	5.1
Abrasion Resistance Index						
Natural rubber = 1,000.						
Bureau of Standards machine	1,200	1,400	800	700	.....	600
Williams machine	2,100	1,300	1,200	1,400	1,200	1,000

bound. Ability of Hycar to absorb energy rapidly is also desirable in vibration dampening service. Flex life is comparable in many instances to that of natural rubber. No broad statement can be made to cover this property.

AGING. Compounds have much greater resistance to effects of oxidation than natural rubber. This quality is outstanding when stocks are operating in service which does not require constant elongation. Resistance to cracking while exposed to light or ozone under tension is superior to that of natural rubber.

ADHESION TO METALS. Stocks can be bonded by vulcanization or cement to most ferrous and non-ferrous metals. Metals must be cleaned and suitably roughened by sand or shot blasting prior to application.

COMPOUNDING AND PROCESSING. Like natural rubber, Hycar is compounded with other ingredients to impart desired properties. It is processed and vulcanized like rubber and can be calendered, molded, extruded and otherwise fabricated on the same machinery as natural rubber.

EBONITE. Hard Hycar has a tensile strength of about 10,000 pounds with elongations from 1 to 5 percent. These compounds have a softening point of approximately 300 deg. F. or about 100 deg. higher than natural rubber. These compounds can be machined readily.

Table II—Oil and Solvent Resistance of Six Compounds

Test	Temperature	Compound Number						Natural Rubber (Tread Stock)
		1	2	3	4	5*	6	
Immersion (48 hours)		Percent Increase in Volume						
Hexane	Room	1.5	0	0	3.0	− 6.1	0	147
Benzol	Room	127	105	86	102	115	76	206
Carbon tetrachloride	Room	37	26	23	37	12	25	262
Acetone	Room	176	152	145	235	192	105	...
Cracked gasoline (X-70)	Room	4.5	1.5	3	4.5	− 9.2	2.2	173
Ethyl alcohol (95%)	Room	9.2	3	3	7.6	−10	3	3
Water	Room	0	0	3	1.5	0.7	0.7	3
Lubricating oil (S.A.E. 20)	Room	0	0	0	0.7	− 6.1	0	30
95% kerosene, 5% benzol	Room	0.7	0	0	1.5	− 4.5	0	153
95% kerosene, 5% benzol	212 F.	11.6	0	0	6.1	−17.4	6.1	483
Light penetrating oil	212 F.	9.2	0	0	5.3	−19.1	4.5	381

* Shrinkages are caused by extraction of softener. This compound was designed for print rolls and is not required to withstand solvents. This type stock may be compounded with other materials to offset any extraction of softener.

Table III—A Rough Guide in Applying Hycar

A — Especially suitable; little or no swelling or softening.
B — Suitable under some conditions, swelling or softening is moderate.
C — Generally unsuitable; swells or softens.

Acetone	C	Ethylene Glycol	A
Acids	See p. 347	Fats, animal	A
Alcohols		Furfural	C
Ethyl	A	Gasoline	A
Amyl	A	Glycerine	A
Benzyl	C	Heptane	A
Alkalis	See p. 347	Hexane	A
Animal oils & fats	A	Kerosene	A
Benzene	B	Mineral oils	A
Butyl acetate	C	Naphtha, petroleum	A
Butane	A	Nitro benzene	C
Carbon Disulphide	B	Octane	A
Carbon Tetrachloride	A	Oils, mineral & vegetable	A
Chloroform	C	Ozone	See p. 347
Cottonseed Oil	A	Phenol	B
Cresol	B	Propane	A
Dibutyl Phthalate	C	Prestone	A
Diethylene Glycol	A	Toluene	B
Dioctyl Phthalate	A	Turpentine	A
Ethyl Acetate	C	Xylene	B

GENERAL PHYSICAL PROPERTIES

Weight. Specific gravity of uncompounded Hycar is 1.0 as compared tc 0.93 for natural rubber.

Color. Black compounds give best physical properties because of carbon black reinforcement. Light colored stocks can be obtained but they do not have quite as good physical properties.

Odor and Taste. Fair. Special compounds are practically odorless and tasteless.

Hardness. Compounds can be produced with a Shore A durometer range from 10 to 100 (Ebonite).

Coefficient of Expansion. 0.00043 to 0.000095 per in. per deg. F.

Electrical Properties. Compounds having high insulation properties and a volume resistivity at 86 deg. F. of 1×10^{11}, as compared to 1×10^{16} for natural rubber, can be produced as well as compounds with resistivity as low or lower than 100 ohms centimeter cube. These are useful in applications involving accumulation of static electricity.

Bloom. Compounds are free from sulphur bloom. A slight bloom is caused on some compounds by anti-oxidant or accelerator chemicals or paraffin.

IMPROVED CERAMICS

A few years ago electrical-mechanical ceramic products could be classed in two groups: porous refractories used in electric heating devices, and porcelains with dense vitrified structures. Recent progress adds several new materials with improved mechanical and electrical properties.

PROPERTIES OF CERAMICS are determined by the physical structure of the finished ceramic, in turn determined by the materials used in the compound and by firing methods. The table compares properties of the more widely used ceramic materials. Greatest improvements have been in dense steatites, rutile compounds, cordierite bodies and, most recently, porcelain.

Steatites are now available in homogeneous and crystalline structures consisting principally of pure clinoenstatite ($MgO.SiO_2$). Base of this material is soapstone or talc, which is plastic and can be worked into intricate shapes. Outstanding properties are high mechanical strength, good electrical properties, and good dielectric strength.

Rutile ceramics consist mainly of the crystalline rutile (TiO_2), have high dielectric constant, are exceptionally hard (8 on Moh's scale) and have good mechanical strength. The material is homogeneous; abrasive wear maintains a highly polished surface. Recent mechanical applications utilize its hardness in place of synthetic jewels and in special alloys for such parts as thread guides.

The rutile bodies are mixed with a small amount of plastic clay for workability. Since clay reduces the properties, it should be kept to a minimum. Hence rutile compounds can only be formed into relatively simple shapes.

Ceramic materials in which the mineral cordierite ($2MgO.2Al_2O_3.5SiO_2$) is the chief constituent, have extremely small thermal expansion. The structure of these bodies, however, is not so homogeneous and the power factor is higher than for steatite or rutile bodies. Outstanding properties are low thermal expansion, good heat shock. Since cordierite materials contain a considerable proportion of clay, they are readily workable.

A new ceramic material is available which is claimed to combine properties of wet-process and dry-process porcelain. It can be molded into intricate shapes. Dielectric strength of this material is about 280 volts per mil, equal to that of the normal wet process porcelain. It has higher mechanical strengths than normal porcelains.

Properties of Ceramic Materials

Materials	Dense Steatites ($MgO.SiO_2$)	Rutile Bodies (TiO_2)	Cordierite Bodies ($2MgO\text{-}2Al_2O_3.5SiO_2$)	Natural Lava	Porcelain	
					Dry Process	Wet Process
Mechanical Properties						
Hardness, Moh's scale	7.5	8	7	6		
Tensile strength, lb./sq. in.	7,500–10,000	7,500	1,500–3,500	2,000–2,500	1,000–2,000	5,000–8,000
Compressive strength, lb./ sq. in.	65,000–85,000	80,000	30,000–40,000	20,000–30,000	30,000–50,000	45,000–90,000
Modulus of rupture, lb./sq. in.	18,000–22,000	20,000	4,000–8,000	8,000–9,000	6,000–8,000	10,000–18,000
Impact resistance, ft. lb./ sq. in. (Charpy)	1.8–2.1	2.4	0.9	1.3–1.4		
Electrical Properties						
Dielectric strength, volts/mil	210–240	100		80–100	40–100	250
Dielectric constant; 60 cycles	6.3–6.5					
1,000 kc	5.8–6.2	85	4.1–5.0	5.3–5.6	6.0–7.0	6.0–7.0
10,000 kc	5.7–6.0	85	4.0–5.0	5.3–5.5	(1 to 1,000 kc.)	(1 to 1,000 kc.)
Power factor, percent; 60 cycles	0.14–0.30				1.70–2.50	1.70–2.50
1,000 kc	0.02–0.20	0.08	0.40	0.30–1.00	0.80–1.00	0.80–1.00
10,000 kc	0.01–0.18	0.06	0.30	0.25–0.90	0.60–1.00	0.60–1.00
Loss factor, percent; 60 cycles	0.88–1.95				10.20–17.50	10.20–17.50
1,000 kc	0.13–1.24		1.64–2.00	1.68–5.30	4.80–7.00	4.80–7.00
10,000 kc	0.06–1.08		1.20–1.50	1.37–4.77	3.60–7.00	3.60–7.00
Volume resistivity, ohms/cc.						
at 25 deg. C. (77 deg. F.)	$>10^{14}$	$>10^{12}$	$>10^{14}$	$>10^{12}$	$>10^{13}$	$>10^{13}$
at 100 deg. C. (212 deg. F.)	$2.1x10^{12}$– $>10^{14}$	$9.8x10^{11}$	$6.1x10^{12}$-$4.0x10^{13}$	$3.2x10^{11}$	(at 75°C.)	(at 75°C.)
at 500 deg. C. (932 deg. F.)	$3.2x10^{5}$–$3.5x10^{10}$	$1.7x10^{6}$	$4.8x10^{6}$–$2.0x10^{7}$	$1.7x10^{6}$–$2.6x10^{8}$		
at 900 deg. C. (1,652 deg. F.)	$<10^{11}$–$2.5x10^{7}$	$<10^{4}$	$7.0x10^{4}$–$2.2x10^{5}$	$2.0x10^{4}$–$1.8x10^{6}$		
Thermal Properties						
Softening point, deg. F	2,550–2,630	3,000	2,570–2,600	2,700–3,000		
Heat resistance, deg. F	1,830	1,830	2,280	2,000–2,190		
Coef. of expansion, per deg. C. (20–100 deg. C.)	$7.8–10.4x10^{-6}$	$8.6–8.7x10^{-6}$	$2.5–2.8x10^{-6}$	$3.4–10.5x10^{-6}$	$3.0–6.0x10^{-6}$	$3.0–6.0x10^{-6}$
Thermal Conductivity Cal. per sec. per cm. per deg. C.	0.006		0.003	0.003–0.005		
General Properties						
Specific gravity	2.6–2.7	4.0	2.1	2.3–2.8	2.3–2.5	2.3–2.5
Weight, lb./cu. in.	0.094–0.098	0.144	0.076	0.085–0.102		
Chemical resistance	Resistant to all common acids and alkalies except hydrofluoric acid					
Water absorption, percent	1.00–0.00	0.05–0.00	8.0–1.0	2.5–1.0	1.0–0.5	0.00

CUPRO-NICKELS

R. A. WILKINS and E. S. BUNN
Revere Copper and Brass Incorporated

THE COPPER-BASE ALLOYS of nickel, known as cupro-nickels, are commercially available in two types: one containing 80 percent copper and 20 percent nickel, and the other 70 percent copper and 30 percent nickel.

Cupro-nickels assumed commercial importance about 1925 following a demand for condenser tubes capable of resisting the erosive-corrosive attack of high velocity salt water used as the coolant in navy surface condensers. Since then 70-30 cupro-nickel has proved superior to 80-20 alloy in condenser tube service. However, the 80-20 alloy continues to be used as condenser tube material for less severe service.

Cupro-nickels are the most resistant of the copper-base alloys to failure by stress-corrosion and corrosion-fatigue. Otherwise the tensile properties of both alloys are similar to those of 70-30 brass.

Both cupro-nickels have been used in strip form for the manufacture of ammunition component parts. Bolts, nuts, screws, and similar parts are manufactured by cold heading operations. Their use is expected to be extended as engineers become better acquainted with the possibilities inherent in these two alloys.

Physical and general mechanical properties of the two cupro-nickels are given in the accompanying charts.

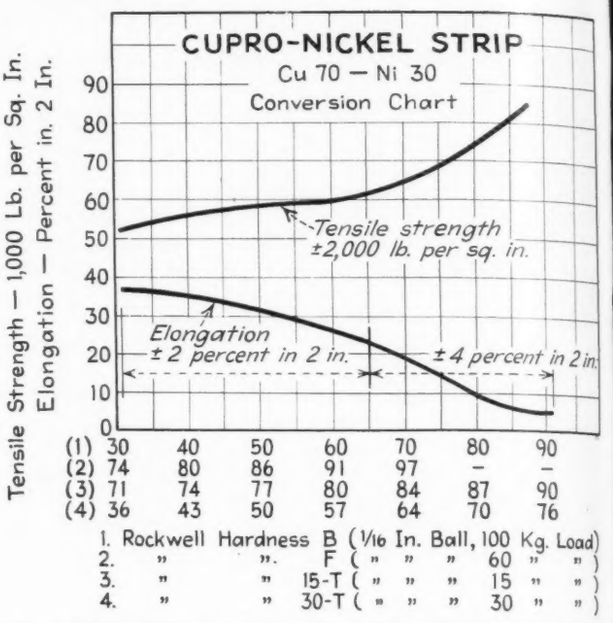

CUPRO-NICKEL STRIP
Cu 70 — Ni 30
Conversion Chart
Tensile Strength — 1,000 Lb. per Sq. In.
Elongation — Percent in. 2 In.
Tensile strength
±2,000 lb. per sq. in.
Elongation
± 2 percent in 2 in.
± 4 percent in 2 in.
(1) 30 40 50 60 70 80 90
(2) 74 80 86 91 97 – –
(3) 71 74 77 80 84 87 90
(4) 36 43 50 57 64 70 76
1. Rockwell Hardness B (1/16 In. Ball, 100 Kg. Load)
2. " " F (" " " 60 " ")
3. " " 15-T (" " " 15 " ")
4. " " 30-T (" " " 30 " ")

Fig. 1—This chart can be employed to determine the approximate tensile strength and percentage elongation of 70-30 cupro nickel when only Rockwell hardness is known. It is accurate for all thicknesses between 0.020 and 0.080 in. within the given limits

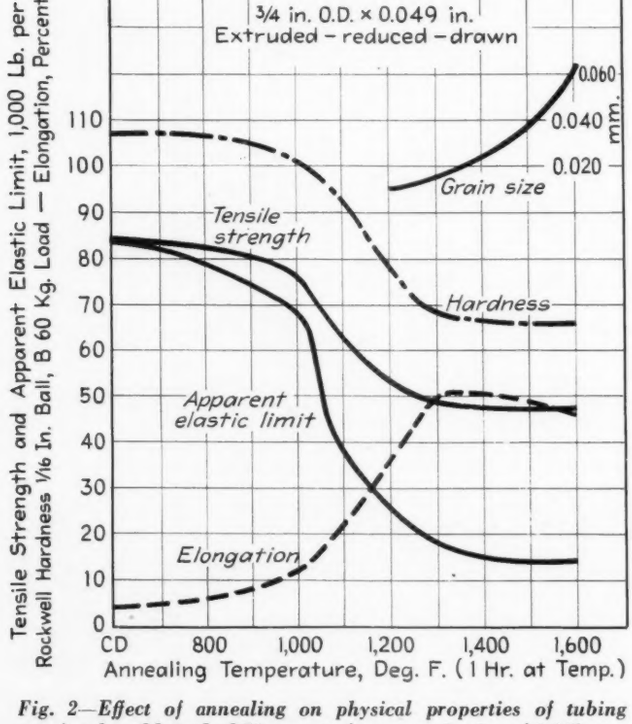

CUPRO-NICKEL CONDENSER TUBING
Cu 70 — Ni 30
3/4 in. O.D. × 0.049 in.
Extruded – reduced – drawn
Tensile Strength and Apparent Elastic Limit, 1,000 Lb. per Sq. In.
Rockwell Hardness 1/16 In. Ball, B 60 Kg. Load — Elongation, Percent in 2 In.
0.060
0.040
0.020
mm.
Grain size
Tensile strength
Hardness
Apparent elastic limit
Elongation
CD 800 1,000 1,200 1,400 1,600
Annealing Temperature, Deg. F. (1 Hr. at Temp.)

Fig. 2—Effect of annealing on physical properties of tubing previously cold worked 70 percent from a grain size of 0.065 mm. (68.94% copper, 30.15% nickel, 0.26% iron, 0.60% manganese)

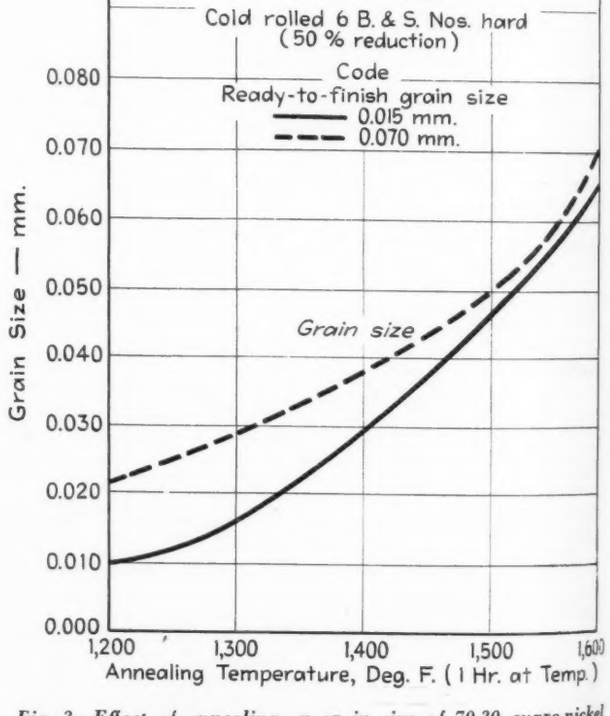


Fig. 3—Effect of annealing on grain size of 70-30 cupro-nickel strip previously cold worked 6 B. & S. numbers hard (50% reduction) from two different grain sizes, 0.015 and 0.070 mm.

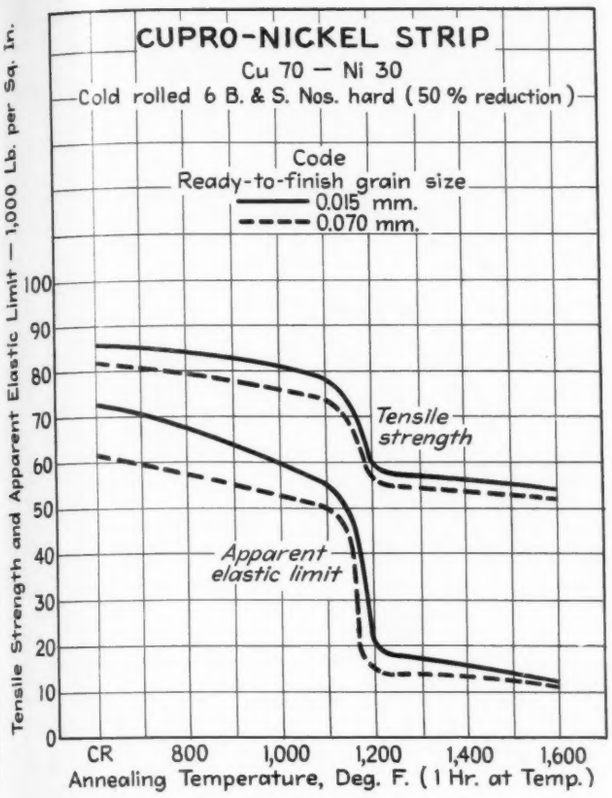

CUPRO-NICKEL STRIP
Cu 70 — Ni 30
Cold rolled 6 B. & S. Nos. hard (50 % reduction)
Code
Ready-to-finish grain size
0.015 mm.
0.070 mm.
Tensile strength
Apparent elastic limit
Tensile Strength and Apparent Elastic Limit — 1,000 Lb. per Sq. In.
Annealing Temperature, Deg. F. (1 Hr. at Temp.)
CR 800 1,000 1,200 1,400 1,600

Fig. 4—Effect of annealing on tensile strength and apparent elastic limit of strip previously cold worked 6 B. & S. numbers hard (50 percent reduction of area) from grain sizes of 0.015 and 0.070 mm.

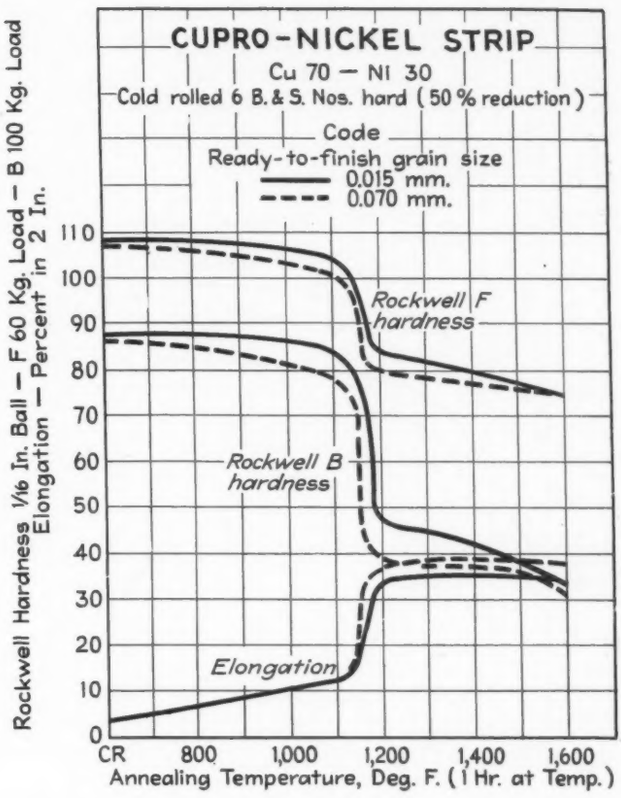

CUPRO-NICKEL STRIP
Cu 70 — NI 30
Cold rolled 6 B. & S. Nos. hard (50 % reduction)
Code
Ready-to-finish grain size
0.015 mm.
0.070 mm.
Rockwell F hardness
Rockwell B hardness
Elongation
Rockwell Hardness 1/16 In. Ball — F 60 Kg. Load — B 100 Kg. Load
Elongation — Percent in 2 In.
Annealing Temperature, Deg. F. (1 Hr. at Temp.)
CR 800 1,000 1,200 1,400 1,600

Fig. 5—Effect of annealing on hardness and percent elongation of strip previously cold worked 6 B. & S. numbers hard (50 percent reduction of area) from grain sizes of 0.015 and 0.070 mm.

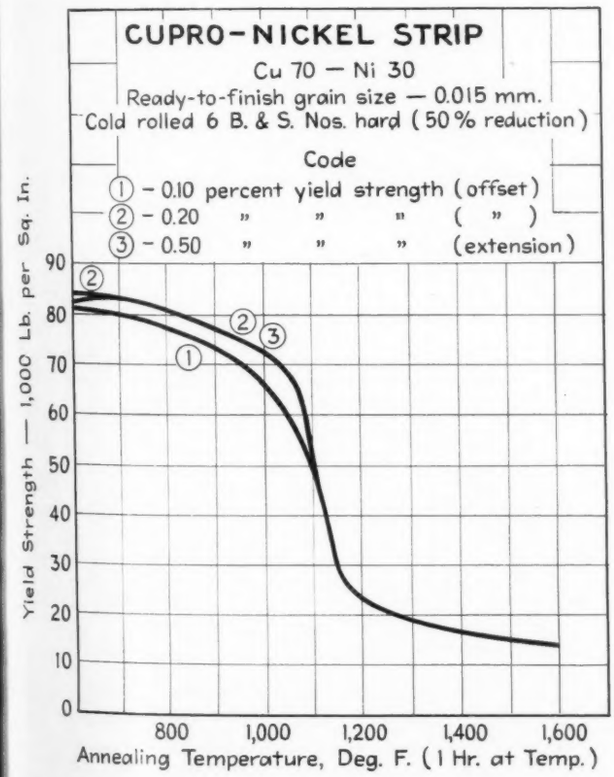

CUPRO-NICKEL STRIP
Cu 70 — Ni 30
Ready-to-finish grain size — 0.015 mm.
Cold rolled 6 B. & S. Nos. hard (50 % reduction)
Code
① - 0.10 percent yield strength (offset)
② - 0.20 " " " (")
③ - 0.50 " " " (extension)
Yield Strength — 1,000 Lb. per Sq. In.
Annealing Temperature, Deg. F. (1 Hr. at Temp.)
800 1,000 1,200 1,400 1,600

Fig. 6—Effect of annealing on yield strength of strip previously cold worked 6 B. & S. numbers hard (50 percent reduction of area) from a grain size of 0.015 mm.

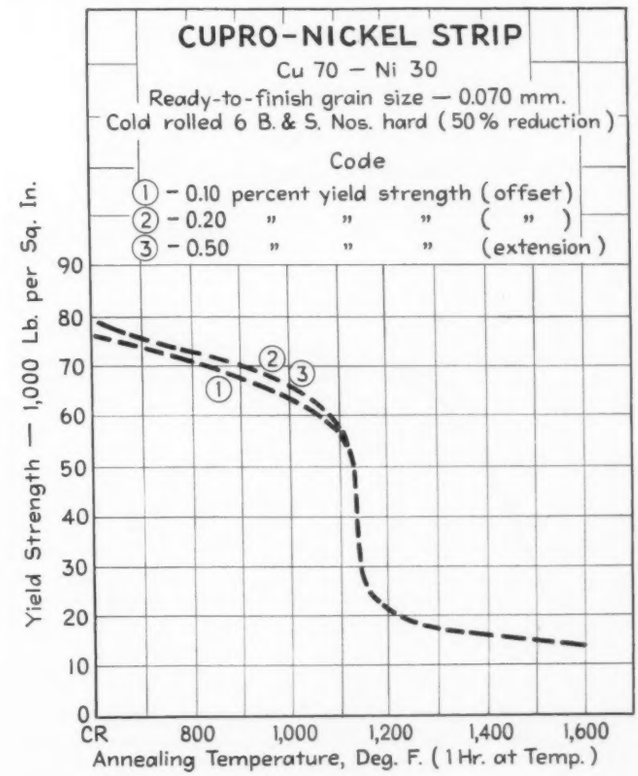

CUPRO-NICKEL STRIP
Cu 70 — Ni 30
Ready-to-finish grain size — 0.070 mm.
Cold rolled 6 B. & S. Nos. hard (50 % reduction)
Code
① - 0.10 percent yield strength (offset)
② - 0.20 " " " (")
③ - 0.50 " " " (extension)
Yield Strength — 1,000 Lb. per Sq. In.
Annealing Temperature, Deg. F. (1 Hr. at Temp.)
CR 800 1,000 1,200 1,400 1,600

Fig. 7—Effect of annealing on yield strength of strip previously cold worked 6 B. & S. numbers hard (50 percent reduction of aera) from a grain size of 0.070 mm.

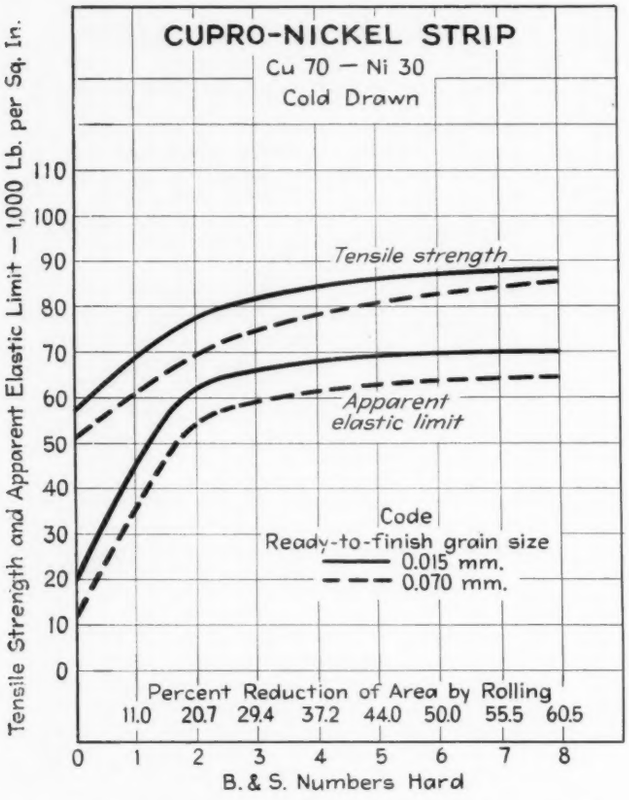

CUPRO-NICKEL STRIP
Cu 70 — Ni 30
Cold Drawn
Tensile strength
Apparent elastic limit
Code
Ready-to-finish grain size
0.015 mm.
0.070 mm.
Tensile Strength and Apparent Elastic Limit — 1,000 Lb. per Sq. In.
Percent Reduction of Area by Rolling
11.0 20.7 29.4 37.2 44.0 50.0 55.5 60.5
B. & S. Numbers Hard

Fig. 8—Effect of cold work on tensile strength and apparent elastic limit of strip (68.94 percent copper, 29.61 percent nickel) previously annealed to grain sizes of 0.015 and 0.070 mm.

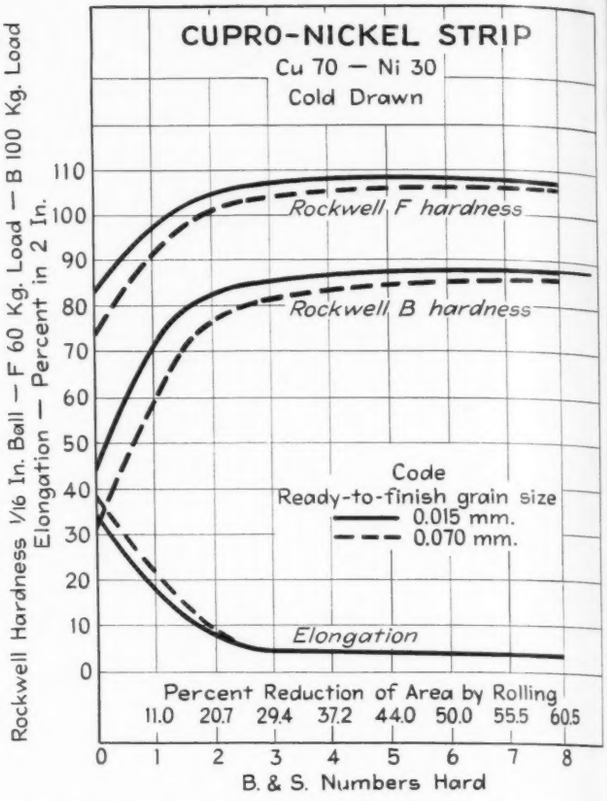

CUPRO-NICKEL STRIP
Cu 70 — Ni 30
Cold Drawn
Rockwell F hardness
Rockwell B hardness
Elongation
Code
Ready-to-finish grain size
0.015 mm.
0.070 mm.
Rockwell Hardness 1/16 In. Ball — F 60 Kg. Load — B 100 Kg. Load
Elongation — Percent in 2 In.
Percent Reduction of Area by Rolling
11.0 20.7 29.4 37.2 44.0 50.0 55.5 60.5
B. & S. Numbers Hard

Fig. 9—Effect of cold work on Rockwell hardness and percent elongation of cupro-nickel strip previously annealed to two different grain sizes, 0.015 and 0.070 mm.

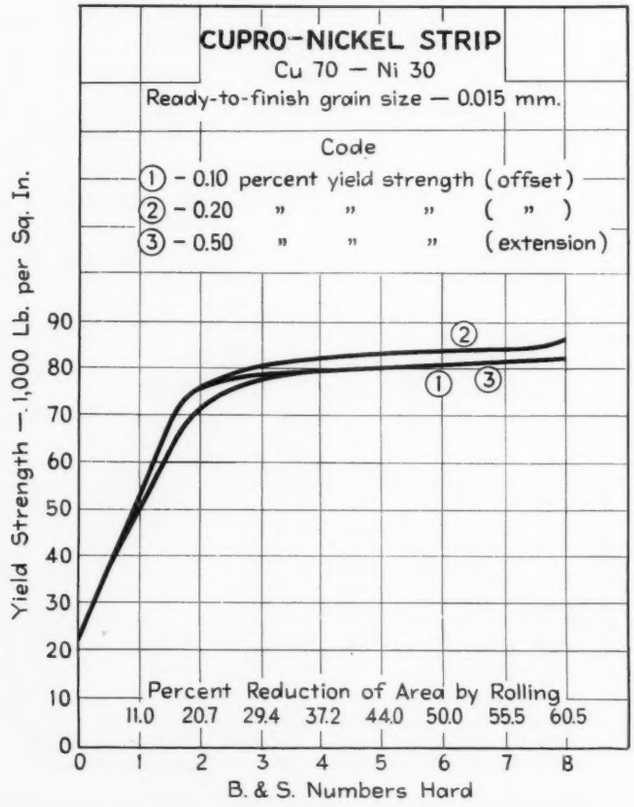

CUPRO-NICKEL STRIP
Cu 70 — Ni 30
Ready-to-finish grain size — 0.015 mm.
Code
① - 0.10 percent yield strength (offset)
② - 0.20 " " " (")
③ - 0.50 " " " (extension)
Yield Strength — 1,000 Lb. per Sq. In.
Percent Reduction of Area by Rolling
11.0 20.7 29.4 37.2 44.0 50.0 55.5 60.5
B. & S. Numbers Hard

Fig. 10—Effect of cold work on yield strengths of strip previously annealed to a grain size of 0.015 mm.

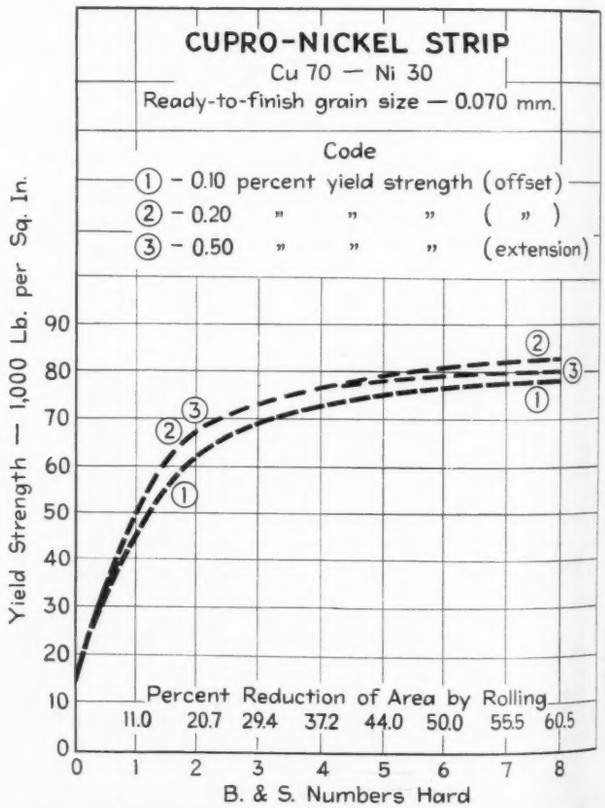

CUPRO-NICKEL STRIP
Cu 70 — Ni 30
Ready-to-finish grain size — 0.070 mm.
Code
① - 0.10 percent yield strength (offset)
② - 0.20 " " " (")
③ - 0.50 " " " (extension)
Yield Strength — 1,000 Lb. per Sq. In.
Percent Reduction of Area by Rolling
11.0 20.7 29.4 37.2 44.0 50.0 55.5 60.5
B. & S. Numbers Hard

Fig. 11—Effect of cold work on yield strengths of strip previously annealed to a grain size of 0.070 mm.

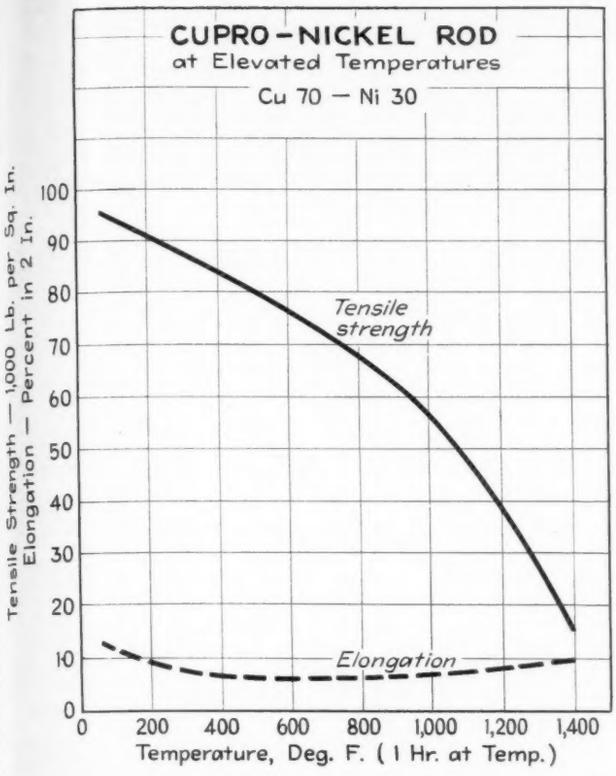

CUPRO-NICKEL ROD
at Elevated Temperatures
Cu 70 — Ni 30
Tensile Strength — 1,000 Lb. per Sq. In.
Elongation — Percent in 2 In.
Tensile strength
Elongation
Temperature, Deg. F. (1 Hr. at Temp.)

Fig. 12—Effect of elevated temperature on tensile strength and percent elongation of rod previously cold drawn to 80 percent reduction of area from material having a grain size of 0.035 mm.

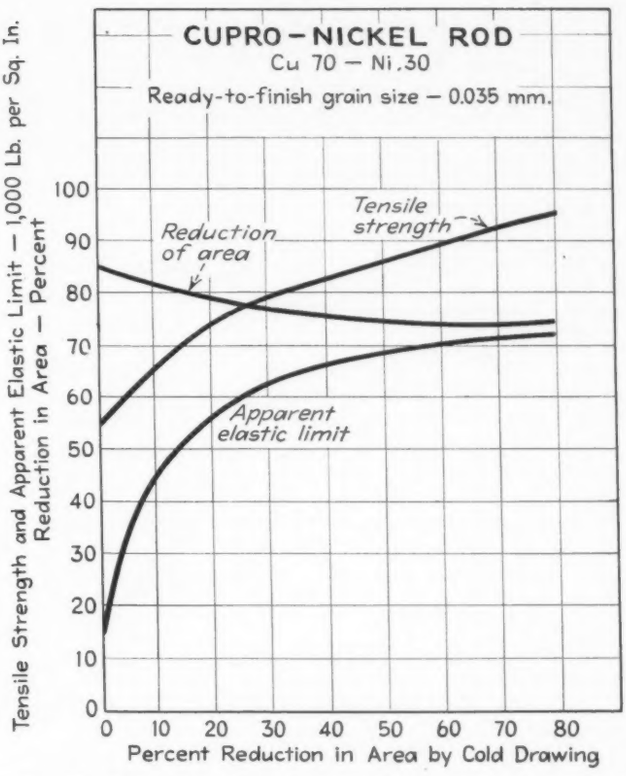

CUPRO-NICKEL ROD
Cu 70 — Ni 30
Ready-to-finish grain size — 0.035 mm.
Tensile Strength and Apparent Elastic Limit — 1,000 Lb. per Sq. In.
Reduction in Area — Percent
Reduction of area
Tensile strength
Apparent elastic limit
Percent Reduction in Area by Cold Drawing

Fig. 13—Effect of cold work on tensile strength and apparent elastic limit of rod (68.56% copper, 30.48% nickel, 0.39% iron, 0.57% manganese) previously annealed to a grain size of 0.035 mm.

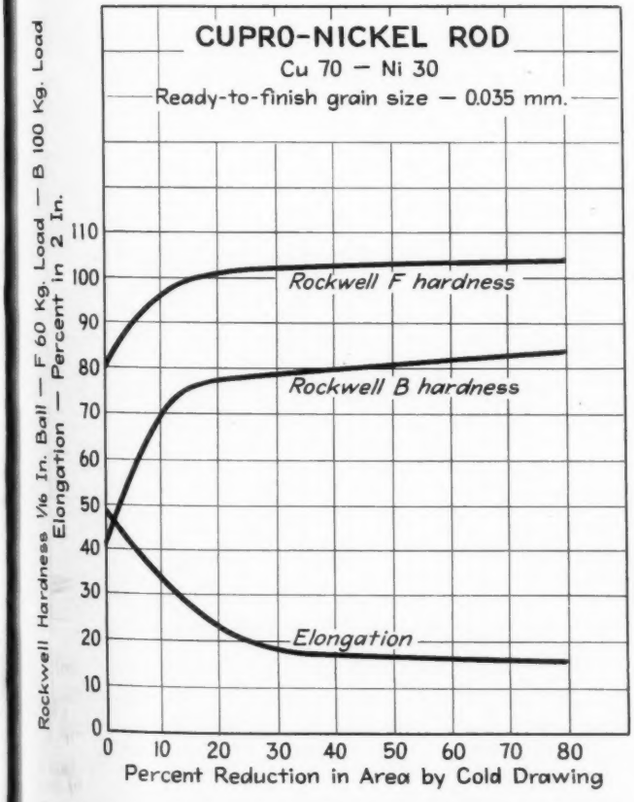

CUPRO-NICKEL ROD
Cu 70 — Ni 30
Ready-to-finish grain size — 0.035 mm.
Rockwell Hardness 1/16 In. Ball — F 60 Kg. Load — B 100 Kg. Load
Elongation — Percent in 2 In.
Rockwell F hardness
Rockwell B hardness
Elongation
Percent Reduction in Area by Cold Drawing

Fig. 14—Effect of cold work on hardness, elongation, and reduction of area of rod previously annealed to a grain size of 0.035 mm.

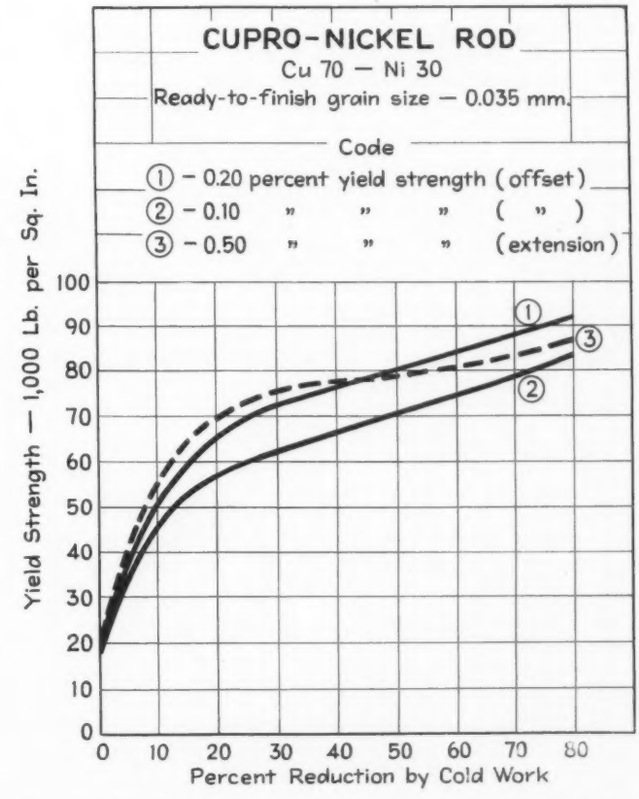

CUPRO-NICKEL ROD
Cu 70 — Ni 30
Ready-to-finish grain size — 0.035 mm.
Code
① — 0.20 percent yield strength (offset)
② — 0.10 " " " (")
③ — 0.50 " " " (extension)
Yield Strength — 1,000 Lb. per Sq. In.
Percent Reduction by Cold Work

Fig. 15—Effect of cold work on yield strength of 70-30 cupro-nickel rod previously annealed to a grain size of 0.035 mm. is shown by three methods of test

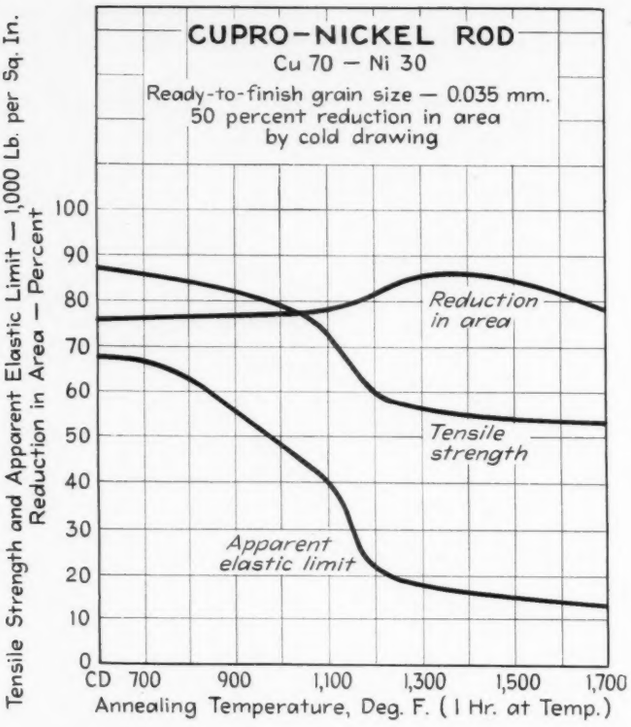

CUPRO-NICKEL ROD
Cu 70 — Ni 30
Ready-to-finish grain size — 0.035 mm.
50 percent reduction in area
by cold drawing
Tensile Strength and Apparent Elastic Limit — 1,000 Lb. per Sq. In.
Reduction in Area — Percent
Reduction in area
Tensile strength
Apparent elastic limit
CD 700 900 1,100 1,300 1,500 1,700
Annealing Temperature, Deg. F. (1 Hr. at Temp.)

Fig. 16—Effect of annealing on tensile strength and apparent elastic limit of rod previously cold drawn to 50 percent reduction of area from material having a grain size of 0.035 mm.

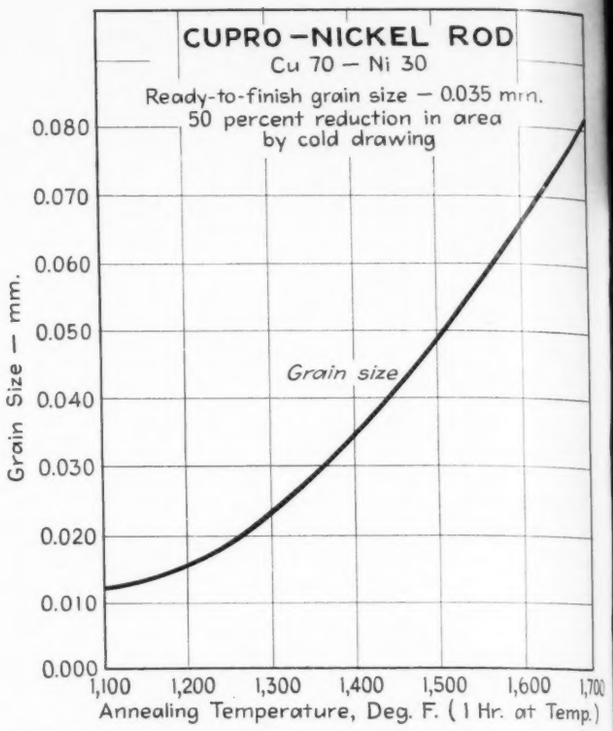

CUPRO-NICKEL ROD
Cu 70 — Ni 30
Ready-to-finish grain size — 0.035 mm.
50 percent reduction in area
by cold drawing
Grain Size — mm.
Grain size
1,100 1,200 1,300 1,400 1,500 1,600 1,700
Annealing Temperature, Deg. F. (1 Hr. at Temp.)

Fig. 17—Effect of annealing on grain size of 70-30 cupro-nickel rod previously cold drawn to 50 percent reduction of area from material having a grain size of 0.035 mm.

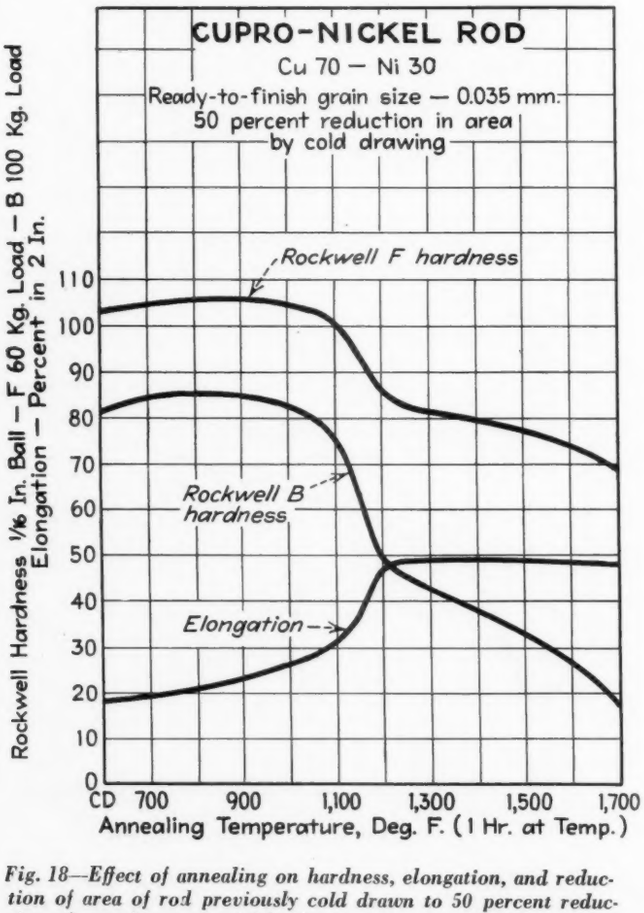

CUPRO-NICKEL ROD
Cu 70 — Ni 30
Ready-to-finish grain size — 0.035 mm.
50 percent reduction in area
by cold drawing
Rockwell Hardness 1/16 In. Ball — F 60 Kg. Load — B 100 Kg. Load
Elongation — Percent in 2 In.
Rockwell F hardness
Rockwell B hardness
Elongation
CD 700 900 1,100 1,300 1,500 1,700
Annealing Temperature, Deg. F. (1 Hr. at Temp.)

Fig. 18—Effect of annealing on hardness, elongation, and reduction of area of rod previously cold drawn to 50 percent reduction of area from material having a grain size of 0.035 mm.

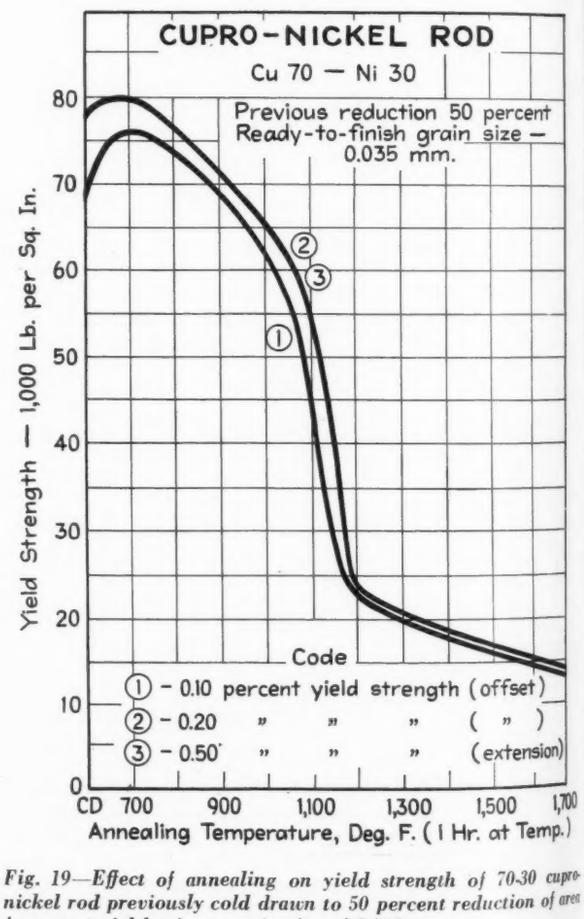

CUPRO-NICKEL ROD
Cu 70 — Ni 30
Previous reduction 50 percent
Ready-to-finish grain size —
0.035 mm.
Yield Strength — 1,000 Lb. per Sq. In.
Code
① — 0.10 percent yield strength (offset)
② — 0.20 " " " (")
③ — 0.50 " " " (extension)
CD 700 900 1,100 1,300 1,500 1,700
Annealing Temperature, Deg. F. (1 Hr. at Temp.)

Fig. 19—Effect of annealing on yield strength of 70-30 cupro-nickel rod previously cold drawn to 50 percent reduction of area from material having a grain size of 0.035 mm.

Question and Comment

Salvage of Aluminum and Magnesium Castings

Aluminum or magnesium castings are often used for parts in which pressure tightness is a prime requisite. The failure of an occasional casting to test pressure tight should not be cause for rejection in most cases if leakage is the result of a very minor unsoundness or shrinkage. Frequently the casting can be salvaged by surface peening, or by impregnation with a sealing material. Of course, the sealing process can be used only where the leakage is slight, and it is not intended to be a substitute for good foundry practice.

Leak Testing

The best leak test is one which exactly duplicates the service conditions. This is often impractical, and a few simple test methods which have been devised will indicate pressure tightness under a wide variety of circumstances. Selection of one test in preference to another will depend on the service requirements as well as the size and shape of the casting.

A test commonly used involves the application of compressed air into the casting after all openings have been covered or plugged. The casting is then covered with a film of soapy water or immersed in water. Usual air pressures used for testing range from 10 to 40 lb. per sq.in. since experience has shown low pressures to be generally as effective as higher pressure.

The most critical test consists of pouring or spraying a film of warm (120 deg. F) soap solution on the casting after it has been heated to between 130 and 150 deg. F. (by immersion in hot water). When air pressure is applied, leaks become apparent as bubbles show on the casting surface. Sensitivity of the soap solution is decreased if castings are at room temperature when tested. Very fine leaks can be detected when the casting is heated but may not show up if the casting is cold.

The immersion method, satisfactory in many cases, is less sensitive than the soap solution method because of the difficulty in observing the escape of air bubbles through water.

Before peening or sealing treatments are initiated, every effort should be made to eliminate leakage by improvements in foundry practice. If, however, due to unusual characteristics of a casting, minor leaks do remain, salvage treatments are often effective. Some very fine leaks can be closed permanently by peening the surface metal toward the leak with a blunt tool in a short stroke air hammer. Blows of the tool should be from the circumference of a small circle around the leak, toward the center, and not directly at the point of leakage. It should be remembered, however, that subsequent sand blasting, polishing or machining may reopen a leak sealed in this way.

Eliminating Severe Leakage

Leaks slightly more severe may be sealed with liquids which will set during air drying or baking. There are several impregnating materials in use at the present time, the commonest of which is sodium silicate solution. Other important ones are double-boiled linseed oil and the synthetic resin varnishes of the glycerol phthalate or phenol formaldehyde type.

A careful selection of the sodium silicate must be made when this material is used since certain grades contain harmful alkalies. A ready-mixed solution of the following composition, having a SiO_2 to Na_2O ratio of approximately 3.5 to 1 is recommended.

Silica	(SiO_2)	27.0 – 29.0%
Sodium Oxide	(Na_2O)	8.0 – 9.0
	(Fe_2O_3)	
	(CaO)	
Impurities	(MgO)	2.0 Max.
	(SO_3)	
	(NaCl)	
Water	(H_2O)	Balance

(Baume Gravity at 65° F. — 39 – 41°)

Note: For the treatment of castings with fine porosity it is necessary to dilute with water to 30° Baume (at 65° F.) for best results.

With large, hollow castings the impregnating operation is usually performed by pouring the sealing material into the heated casting. Prior to this, the smaller core openings in the casting should be closed by means of expanding rubber plugs and the larger openings by rubber-faced covers held in place with C clamps of suitable size. One opening, or a group of small openings, should be left for the introduction of the impregnating material and the attachment of an air line through which pressure can be applied.

The casting should be heated to approximately 200 deg. F. An open-end steam line directed on the casting is a simple way of doing this, and blanketing the casting with several thicknesses of canvas will help to hold the temperature. The impregnating material should be introduced slowly into the hot casting to release entrapped air.

After the casting is filled, the air line should be attached and pressure applied for a sufficient length of time to insure penetration of the sealing material into the pores of the casting. An air pressure of 60 lb. per sq.in. is usually adequate, but both the pressure and length of time to apply it should be determined experimentally for each type of casting.

In place of this treatment, and where smaller size permits, castings can be immersed in the impregnating solution at 150 to 200 deg. F. for a period of four or more hours. At the end of this time they should be removed, washed in water (in the case of castings impregnated with sodium silicate) and air dried or baked.

It is not always economical to treat small castings individually by the internal pressure method. Also, many castings have no hollow cavity. In the batch pressure method a number of such castings can be immersed in the sealing solution. Pressure is then applied to the surface of the solution. The tank in which batch pressure sealing is performed may be heated by a steam coil, and the cover sealed on after being locked in place by a self-closing rubber gasket. An air line to supply the necessary pressure leads into the tank below the level of the solution. The usual run of foundry castings may be satisfactorily impregnated over a six-hour run at 50 lb. pressure.

Draining and Baking

After the impregnating process, the solution should be drained out thoroughly so that only a uniform coating remains on the surfaces of the casting and none is trapped in recesses. A coating that is too heavy, or pools of the solution remaining inside, will not dry

thoroughly and will remain, instead, as an undesirable, gummy mass. Castings can be drained on a suitable rack.

During the baking process, the castings should be packed in a position which will allow any impregnating solution softened by the heat to drain out. If one side of the casting shows more tendency to leak than another, that side should be placed downward to allow longer soaking in the solution.

Any suitably controlled oven is satisfactory for baking. Temperature and time limits for baking depend on the size of the casting and the type of impregnating material used. They should be gauged so as to produce a thoroughly baked coating without carbonizing. Synthetic resin varnishes require lower temperatures than linseed oil—a fact that must be kept in mind to avoid destruction of the film by too high a temperature. Manufacturers of synthetic resin varnishes usually specify the temperature to be used with their products.

The following baking temperatures have been used successfully:

Sodium silicate solution
1 hr. at 215–350 deg. F.
Double boiled linseed oil
3 to 5 hr. at 500–550 deg. F.
or 20 hr. at 440 deg. F.
Glycerol phthalate resin varnishes and phenol formaldehyde resin varnishes
1 to 2 hr. at 275–320 deg. F.

When used with heat-treated alloys, some minor changes may be necessary or desirable in order to incorporate these impregnating treatments with standard heat-treating practice. Such changes should be considered for each particular case.

[Editor's Note: For further information on this subject see article "Salvage Specifications" on page 78 in the Feb. issue of Product Engineering.]

Wants Data On Rate of Flow

To the Editor:

In the design of a gas mixing valve for internal combustion engines, large clearances are desirable for manufacturing reasons and to insure freedom for sensitive governing. Gas leakage at low throttle, up to some point, will have a desirable enriching effect but beyond that point will reduce power and waste gas.

I can find plenty of data on flow of gases through long pipes, or the Simplon tunnel, but nothing on leakage through a 0.001 in. crack ½ in. in depth. I do not feel that it is safe to apply nozzle formula to this quite different order of flow.

Do you know of any experimental data on this matter of rate of flow through capillary passages? If not, it would seem a good subject for an article.

A short taper plug centered in a long B & S taper reamed hole and adjustable axially would do the trick and give a fair range of sizes while different depths of crack could be had by cutting off the plug. The slight difference in area between the large and small ends of the plug would offer a chance to work in some fancy math. —F. D. Howe
Columbiana, Ohio

[Editor's Note: We have no such information on hand and would like to see something on the subject.]

Don't Go to Washington

To the Editor:

An illustrated booklet titled, "How Small Industries Can Go After War Work," has been published by this association to show small companies how to go after contracts or sub-contracts through the Contract Distribution Branch, Production Division, War Production Board, Washington, D. C., or through its Regional Offices in many cities in each state.

In its foreword the booklet says, "Because of the unusually large number of persons who go to Washington, the personnel of the Contract Distribution Office there is taxed beyond its facilities. Officials of small industries are, therefore, requested to contact the Regional Office nearest the location of their plant. The directors and personnel of these offices are as well versed in giving information as to how contracts can be obtained as the main office in Washington. It is the purpose of the Contract Distribution Branch to harness the full productive power of American Industry in this titanic War Armament Program. Every manufacturer, large and small, can serve his country and himself by making full use of this service."

A list of Regional Offices and addresses, names of officers and telephone numbers are included.

Further information tells how industry can secure loans for equipment to make war armament, how small companies can form pools to obtain contracts and other useful data.

A copy can be obtained without charge by writing to this association. —Copper & Brass Research Assn.
New York, N. Y.

Can You Work This One?

H. E. SMITH

Solution to May problem—

Profit and Loss

Since the gain on one horse was $300, $(x + .25\,x = 1500)$, and the loss on the other was $500, $(x - .25\,x = 1500)$, the loss on the transaction was $200. A loss of 15% $(x\ 2000 = 300)$ would still allow an even break.

This month's problem—

Beggars' Game

After a busy day, five beggars sat down in a circle, each with his heap of pennies before him. All five heaps were equal.

Then spoke the eldest and wisest beggar, unfolding, as he spoke, an empty sack, "My friends, let me teach you a pretty little game! First, I name myself 'Number One', my left-hand neighbor 'Number Two', and so on to 'Number Five'. I then pour into this sack the whole of my earnings for the day and hand it to him who sits next but one on my left, that is 'Number Three'. His part in the game is to take out of it and give to his neighbors as many pennies as represent their names (that is, he must give four to 'Number Four' and two to 'Number Two'.) He must then put into the sack half as much as it contained when he received it; and he must then hand it on just as I did, that is, he must hand it to him who sits next but one on his left—who will, of course, be 'Number Five'. He must proceed in the same way, and hand it on to 'Number Two', from whom the sack will find its way to 'Number Four', and so on to me again. If any player cannot furnish, from his own heap, the whole of what he has to put into the sack, he is at liberty to draw upon any of the other heaps, except my heap"!

The other beggars entered into the game with much enthusiasm, and in due course, the sack returned to 'Number One'. He put into it the two pennies he had received during the game and carefully tied up the mouth of it with a string. Then, remarking, "It is a very pretty little game," he rose to his feet and hastily quitted the spot. The other four beggars gazed at each other with rueful countenances. Not one of them had a penny left.

How much had each at first?

News and Summaries

Emergency Specifications By A.S.T.M. Committees

TECHNICAL COMMITTEES of the American Society for Testing Materials which have developed a number of changes and new specifications to expedite the procurement and conservation of strategic materials during the emergency period are: A-1 on steel, A-5 on corrosion of iron and steel, A-10 on iron-chromium-nickel alloys, B-1 on copper wires for electrical conductors, B-2 on non-ferrous metals and alloys, B-5 on copper and copper alloys, B-6 on die-cast metals and alloys, D-1 on paint, varnish and lacquer, D-11 on rubber products, and D-12 on soaps and other detergents.

Five complete emergency standards recently issued by the Society cover lead coated and lead-alloy coated copper wire for electrical purposes, hot-dip lead coating on hardware, method for conducting salt spray tests on organic protective coatings, hardness conversion table for cartridge brass, and carbon-chromium ball and roller bearing steels. Emergency provisions provided in over 42 standards can be used by the purchaser as a guide to alternates for specific applications or uses.

To place emergency actions in effect quickly, A.S.T.M. procedure calls for approval in subcommittee, endorsement of main standing committee chairman, and approval by the Committee E-10 on standards. The latter consists of six outstanding authorities who must decide whether a substantial consensus has been reached.

Method Discovered For Iron-Glass Seal

TIGHT SEALS between iron and glass, long a problem in the manufacture of electric lamps and radio tubes, are now possible through a method discovered by Drs. Albert W. Hull and Louis Navias of the General Electric Research Laboratory. An important result, particularly at this time, is the elimination of the need for nickel and cobalt in wires usually required for high power vacuum tubes.

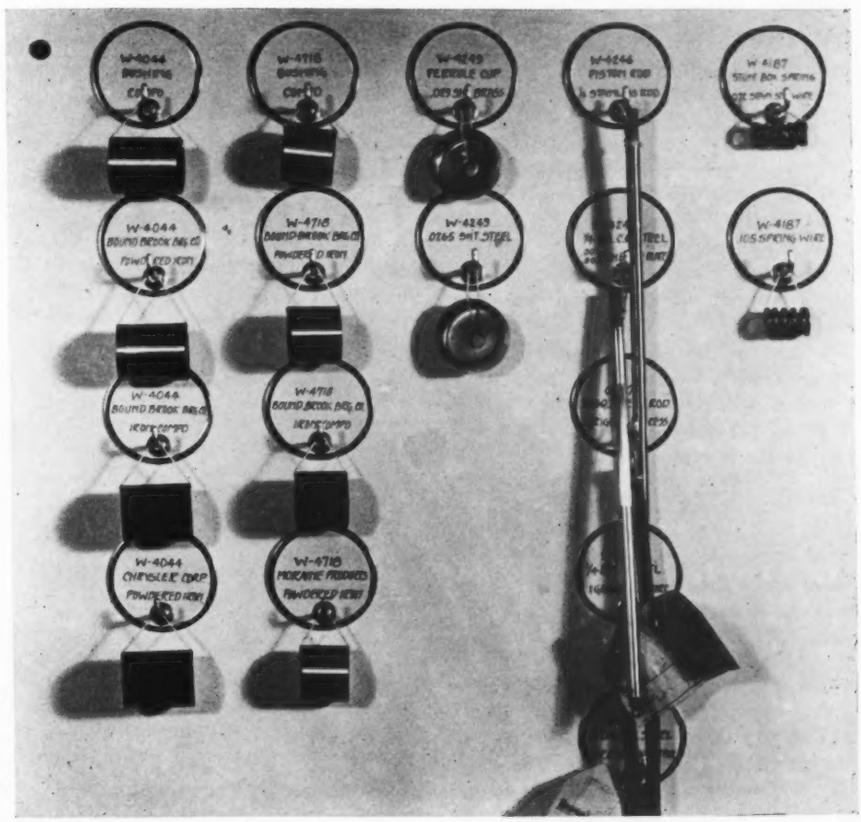
The difficulty has been in finding a metal-glass combination in which both elements have the same heat expansion characteristics. Platinum was first used, and various substitutes were devised, but although they were satisfactory for light bulb seals, they were not suitable for adaptation to large seals of powerful vacuum tubes.

Among a series of glasses having a rate of expansion very close to that of iron, is one recently developed of the following composition: 45 percent silicon dioxide, 14 percent potassium oxide, 6.0 percent sodium oxide, 30 percent lead oxide, and 5.0 percent calcium fluoride.

Another invention of Dr. Navias is an air tight seal for a joint between lead bearing glass and iron wire. When such a joint is sealed, lead atoms migrate from the glass to the metal, causing a weak joint which may let in air. In the new procedure, lead-free glass is placed directly over the metal, and the lead-containing glass sealed directly to the first glass layer. The latter prevents migration of lead atoms and is thin enough not to crack and let in air.

Copper-Silver Alloys

ADVANTAGES in adding a small percentage of silver to copper are reported by the American Silver Producers' Research Project. Silver is said to be es-



One of two boards on the wall of the engineering department of Dayton Pump & Manufacturing Company to systematize substitutions of materials on water pumps and to eliminate confusion. Not all of these changes have been made in production, however, since substitution was studied with preparedness in mind. Factors considered in making substitutions in pumps are: (1) functional interchangeability, based on strength and resistance to wear and corrosion; (2) service interchangeability; (3) customer reaction from service standpoint; (4) adaptation to current production methods. Second factor is the most difficult because material changes often necessitate design changes, and such changes should not prevent parts replacement in service.

pecially useful in copper which is to be soldered, for example, commutator bars, as the alloy formed is not unduly softened by the heat applied.

The report says further that fractional percentages of silver in photo engravers' plates prevent the softening of cold-rolled copper during processing at elevated temperatures. Coppersmiths are reported to favor a copper welding rod containing about 1.0 percent silver and a trace of deoxidizer. Some copper alloys designed for high electrical conductivity may contain 1.0 percent silver, over 95 percent copper, and hardening elements such as beryllium, cobalt, cadmium or tin. In brazing alloys, up to 85 percent copper, silver in amounts of 10 percent or more is commonly used. Principal advantages in the use of silver in such alloys is increased fluidity, good penetration into small clearances, and low melting point.

Plating Zinc on Steel From Waste Materials

USING WASTE instead of new metals, a new process will interest manufacturers of wire, strip and other metal products. It is the Hubbell-Weisberg Process of electroplating steel with zinc. Its advantage is said to lie in its ability to use cheap sources of zinc instead of zinc anodes. Zinc is dissolved from galvanizer's skimmings, brass foundry fume, organic reduction residue or ore by a solution of ammonium chloride and ammonia. Insoluble graphite anodes are used in the plating tank. Plating reduces the zinc content of the plating solution which is then returned to the leaching plant where the zinc removed by plating is replaced.

The process produces zinc coated wire or strip which, it is claimed, is equal to that produced by other zinc plating methods. Steel handling, pickling and cleaning are the same as for other processes. The plating tank has been specially designed to take advantage of the fact that anodes seldom have to be changed. It provides covers which prevent the escape of ammonia fumes, and at the same time allows easy replacement of wires or strips. The outstanding value of the process is said to lie in the savings resulting from the low cost of the zinc going into the plating operation. These savings increase as the amount of zinc used increases because a special plant has to be provided to put into solution zinc in skimmings or other products, and the operating costs of this plant must be added to the purchase price of zinc bearing scrap to obtain the actual cost of the zinc used in the plating tank. The process becomes economical when about 1½ to 2½ tons of zinc are required every twenty-four hours, depending upon the cost of zinc materials available. As the daily consumption of zinc increases above this minimum, the savings become increasingly impressive.

Savings possible by the use of the Hubbell-Weisberg Process have been estimated at as much as 3½ cents per lb. of zinc deposited, as compared with furnishing zinc in the form of high grade anodes at 8¼ cents per pound.

Meetings

American Society for Testing Materials—Annual Meeting, Chalfonte-Haddon Hall, Atlantic City, N. J., June 22-26.

American Society of Mechanical Engineers — Semi-annual Meeting, Hotel Statler, Cleveland, Ohio, June 8-11.

American Institute of Electrical Engineers—Summer Convention, Drake Hotel, Chicago, Ill., June 22-28.

High-Grade Chromium From Chromite Ore

A PROCESS for the conversion of chromite concentrates into a higher grade of material which may be used to produce either high-purity chromium or standard ferrochromium for the manufacturer of alloy steels has been announced by Dr. R. R. Sayers of the Bureau of Mines. It is expected that the new method, a roasting and leaching process, will help satisfy urgent wartime demands for chromium, making us less dependent on foreign sources. The ratio of chromium to iron in the ore can be raised from about 1.7 to 1 up to as high as 30 or 40 to 1 by this process.

According to the Bureau of Mines and the Geological Survey, deposits of low-grade Chromite ore in Montana, alone estimated at 2½ million tons, are believed to contain enough chromium to yield 900,000 tons of chromite (concentrates containing 45 percent or more chromic oxide) of a grade suitable for making ferrochromium. This is equivalent to 4½ times the chromite from domestic mines in the past 28 years.

Strength of Plastic Threads

IN A REPORT from the Plastics Institute Research Project X-129, A. Champagne and E. Lofgren announced results of tests on strength of threaded sections in plastic materials.

Plastics tested were 6-in. rods of ½-in. diameter acrylic and cast phenolic and a 5x8x½ in. block of laminated canvas phenolic. Equipment used were tap and die sets N.C. ½ in., 13 threads per in.; 96 F. ½ in., 20 threads per in.; and a Tinius Olsen testing machine.

After threads had been formed in the rods, they were mounted in the jaws of the testing machine. Tension was applied with the aid of a steel hex nut. The block of laminated canvas plastic having been drilled and tapped, a 6-in. machine bolt was inserted and tension applied to the threads.

Results of the tests are as follows:

Plastic Material	Failing load in lb.	Thread Detail
Cast phenolic	620 445 550 1,120	½ in., 20 threads per in. Test length—⅜ in.
	Ave. 682	
Acrylic	585 670 450 570	½ in , 20 threads per in. Test length—⅜ in.
	Ave. 569	
Acrylic	550 360 250 300	½ in., 13 threads per in. Test length—⅜ in.
	Ave. 365	
Laminate phenolic	3,810 5,600 5,480 6,260	½ in., 20 threads per in. Test length—½ in.
	Ave. 5,287	
Laminate phenolic	4,260 3,520 4,820 3,820	½ in., 13 threads per in. Test length—½ in.
	Ave. 3,805	

Do You Know That—

BRAZING is now being done with high frequency heating equipment. Advantages: localized heating, no controlled atmosphere necessary, minimum distortion, clean joints, close quality control. (8)

CHIPS from the production of 75 and 105 mm. high explosive shells at a General Motors plant weigh some 80 tons a day. They are removed by an underfloor conveyor. (9)

A new normally transparent glass becomes opaque when viewed at a 45 deg. angle. It has possibilities as a means of protection of plants from night bombers. (10)

ENGINEERING IN WASHINGTON

McGRAW-HILL WASHINGTON BUREAU, *PAUL WOOTON, CHIEF*

Combat airplane development is going ahead faster than ever, even with the industry burdened with concentration on mass production of existing types. The industry now has 20 new types of planes in development. The problem of making equipment for air war is a complete paradox: you can't beat the enemy unless you freeze design long enough to build more planes than he has, and you can't beat him unless you keep ahead of him in the design of better planes.

40,000 tons of silver, over four-fifths of the Treasury's uncoined supply of the metal, will be used as a substitute for tin in solder and for copper in bus bars used to conduct heavy loads of electrical power. The Treasury is drafting legislation to make newly mined domestic silver available for industrial users, at the world market price of about 35 cents. Stainless steel, nickel and tin are saved by substituting silver plated cutlery for the Army. Formerly a base coating of nickel and copper was deemed necessary under silver plating. New developments make it possible to apply the silver directly to the base metal with only a slight increase in the amount of silver.

Germany is reported working on bombers with pressurized cabins. This might well be so, because the battle for altitude is getting hotter. All the oxygen systems in use are a serious handicap to the crews. Bombing planes can already go so high that AA guns are ineffective, and if they can get higher than the enemy can fly they can bomb with impunity. Incidentally, all Japanese planes are so weak on altitude that when the United Nations get there in force, the Nippons will be in serious trouble.

At the site of Shasta Dam seven cableways radiate from this 460 ft. head tower to movable tail towers, three of which are seen in the distance. On these cableways, 8 yd. buckets of concrete are carried to desired locations. Tail towers are moved on tracks. Cement aggregates are carried on a 10-mile conveyor belt system to the cement mixing plant at the right of the head tower. Tail towers and conveyor system are powered by Westinghouse gearmotors.

Germany also appears to be working up a light-tank-carrying plane with the rear section of the fusilage swinging around on hinges to form an open-end hatchway.

The Sea Otter has come up to be shot at again. Because merchant ship construction is seriously behind the need for bottoms to carry the increasing deluge of war supplies abroad, Navy has turned over the 300 ft. cargo vessel to another manufacturer for redesign. You may recall that the original was propelled by 12 or 16 automobile engines, geared to a vertical shaft that extended vertically down through the bottom to a propeller amidship. Presumably the intention of the designers was to save the steel and forge work on a convention long shaft to the stern. One of the main troubles was that the mid-ship propeller increased the draft of the Otter, which was supposed to be a shallow-draft boat. Intended changes are not announced, but opinion is that the propeller will go on the stern. Reports are the gasoline auto engines will be used for dry cargo, diesel for oil and inflammable goods.

Hot machine guns will be handled by soldiers with asbestos mittens re-designed to save about 300,000 pounds of asbestos cloth this year. A new type of heat-resistant interlining has been substituted for asbestos.

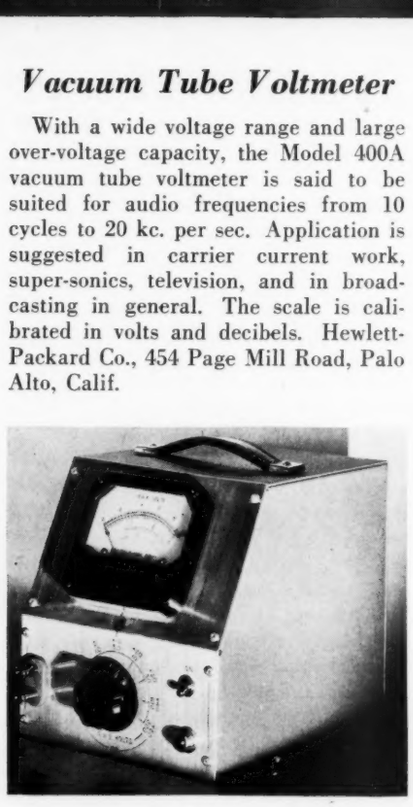
A new process to produce chromium, developed by the U. S. Bureau of Mines, may prevent shortages of chromium in the war industries. Beach sands from Coos County, Ore., and chromite ore deposits near Columbus, Mont., have been combined to produce the metal, according to Dr. R. S. Dean, Chief Engineer of the Bureau's Metallurgical Division. The new chemical process supplements present methods of concentrating ore and sands and is expected to improve the quality of domestic chromium.

About 20,000 pilot's charts printed by the Navy Hydrographic Office will be rolled up in small watertight metal tubes and stowed in lifeboats of Allied ships. Navy reports that anyone cast adrift in an open boat can figure his course from the charts if he knows his approximate position.

New Materials and Parts

Vacuum Tube Voltmeter

With a wide voltage range and large over-voltage capacity, the Model 400A vacuum tube voltmeter is said to be suited for audio frequencies from 10 cycles to 20 kc. per sec. Application is suggested in carrier current work, super-sonics, television, and in broadcasting in general. The scale is calibrated in volts and decibels. Hewlett-Packard Co., 454 Page Mill Road, Palo Alto, Calif.

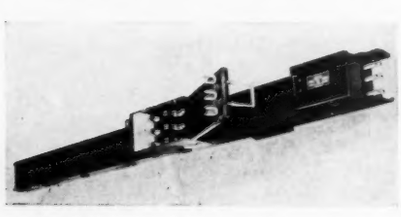


Laminated Insulation

Produced in the form of laminated insulating sheets, tubes and rods. Parts machined from these forms are said to show ten times the arc resistance formerly available in laminated insulating materials. They are provided in all the usual laminated grades—paper base, canvas base, muslin base, and fibre glass base. Formica Insulation Co., Cincinnati, Ohio.

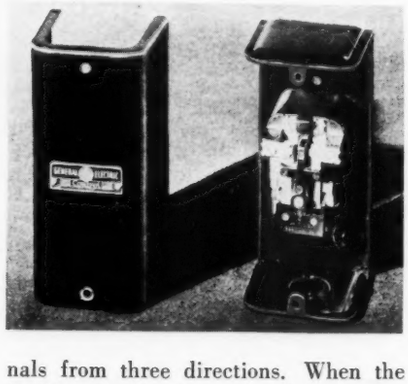
Plug-In Type Duct

Accessibility and flexibility are claimed for this "Saflex" plug-in duct. Available in capacities of 250, 375, 500, 750 and 1,000 amp. in 2 and 3-wire single phase, 3 phase, and 4 wire service. It utilizes round tubular buses for 250, 375 and 750 amp. capacity and solid, round bus bar for 500 and 1,000 amp. Bus bars are contained in a steel housing produced in standard 10-ft. lengths. Six openings on each side of a 10-ft. length will accommodate 12 plug-in units. Square D Co., 6050 Rivard St., Detroit, Mich.



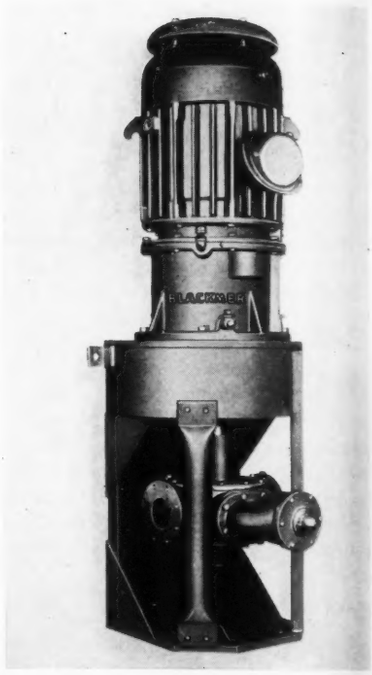
Magnetic Relay-Contactor

For use as a relaying unit or motor starter. The magnetic relay-contactor comes in three contact arrangements—single-pole normally open, double-pole normally open, and double-pole, double-throw. The CR 2790 can be supplied either open or inclosed in a general purpose housing. On the inclosed type, wiring and servicing are simplified through use of a U-shaped cover, allowing access to the front connected terminals from three directions. When the double-pole form is used with a momentary contact, push-button station, under voltage protection can be provided by using one pole as a holding interlock in the coil circuit. As a relay, the unit is rated at ten amp. As a motor starter, the normally open tips are rated ½ hp., 110 volts a.c., and 1 hp., 220 volts a.c. Case measures 3 x 2¾ x 6⅕ in. General Electric Co., Schenectady, N. Y.



Vertically Mounted Rotary Pump

With capacity of 150 g.p.m. at pressure of 100 lb. per sq. in., this pump works on a "bucket" or swinging vane principle, handling a wide range of liquids with viscosities ranging from that of tar to naphtha. Self-adjusting for wear, the pump is claimed to have no loss of volumetric efficiency during the life of the buckets. When the latter are worn out, the unit may be restored to original efficiency by replacing old buckets with new. The same type of pump and mounting is also furnished in capacities from 5 to 700 g.p.m. at pressures up to 300 lb. per sq. in. for marine and heavier industrial applications. With the small space needed for mounting, the pump is intended for applications where floor space is at a minimum. Blackmer Pump Co., Grand Rapids, Mich.

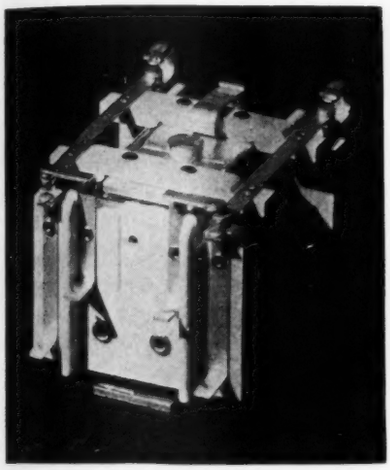


Metal Finish

When applied on polished metals, a colored transparent finish makes them resemble copper, brass, bronze, color-treated aluminum and steel, or other metals. Known as "Plate lustre," the finish consists of a concentrated enamel of the desired color added to a clear lacquer. Applied by spraying or roller coating, it can be furnished for either air drying or baking. Flat stock finished with certain grades can be formed after baking. Two-color finishes can be obtained by "relieving" operation, and colors are said to be fast to light in indoor service. Maas & Waldstein Co., Newark, N. J.

Relays for Aircraft

Three types of lightweight compact relays. (1) Single-pole relay comes in two forms, a single-circuit with one normally open contact, and a two-circuit with one normally open and one normally closed contact. Maximum continuous current rating is 20 amp. at 12 or 24 volts. Maximum make or break rating is 100 amp. at 12 or 24 volts. (2) High-voltage relay for aircraft radio transmitters, illustrated. Contacts arranged for double-pole, double-throw, double-break operation, have maximum rating of 0.020 amp. at 1,000 volts d.c., and 0.100 amp. at 500 volts d.c. (3) Two and three-pole relays in two forms, one has normally open circuit per pole and the other normally closed circuit per pole. Maximum continuous current ratings are 8 amp. at 12 or 24 volts. Maximum make or break



ratings 26 amp. at 12 or 24 volts. For all types coils operate at 1.2 watts. All are for use in an ambient temperature from −40 deg. to +95 deg. C. They can be used at rated current at altitudes to 40,000 ft. Mechanical frequencies allowed are 5 to 55 cycles per sec. at 3/32 in. maximum amplitude. Contacts remain in correct position when subjected to linear accelerations of 10 times gravity in any direction. General Electric Co., Schenectady, N. Y.

Protective Coating

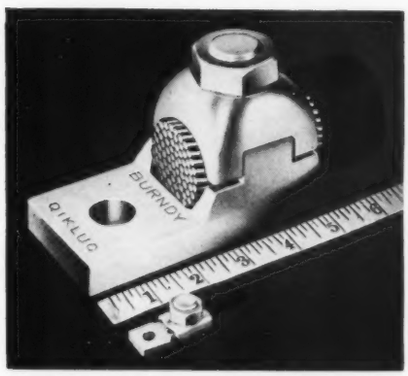
"Chempruf," a coating claimed to be resistant to all acids and all alkalies in any concentrations and at any working temperatures, comes in two types, "A" and "B." The former is a brush-on heavy liquid that can be applied by any workman; the latter is a plastic applied only by the manufacturer's representative in their plant or on location. Softening point for the material is 300 deg. F.; melting point, 510 deg. F. Where mechanical abuse is not a consideration, four coats of Chempruf A is sufficient. Chempruf B may be of any thickness from 1/8 in. Manufacturers state that this coating is not to be used with mineral solvents. Protective Coatings, Inc., 10391 Northlawn, Detroit, Mich.

Chemical Black for Copper

Suitable for blackening copper, beryllium copper, bronze, phosphor bronze and brasses with zinc content of 35 percent or less. Called Ebonol "C," it is a blackening salt effective at temperatures from 200 to 212 deg. F. in from 2 to 10 min., depending upon the alloy used. Claims are that the coating, essentially cupric oxide, cannot chip or flake, is hard enough to be buffed, and does not require lacquering to prevent wear. A part so finished is said to withstand temperatures as high as 700 deg. F. without harm. Non-copper alloys which can be copper plated can thus be blackened by treating the plating. The Enthone Co., 442 Elm St., New Haven, Conn.

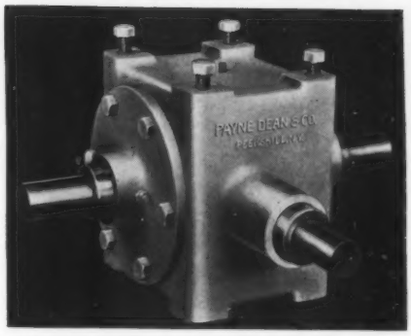
Cable Terminal Connections

Covering a wide range of conductor sizes, a new terminal connector "Qiklug," comes in eleven sizes from No. 14 solid to 2,000,000 cir. mils. As many as twelve different sizes of cable can be accommodated by some sizes. All but the very small connectors are provided with serrations on cable clamping elements. Burndy Engineering Co., Inc., 107 Eastern Blvd., New York, N. Y.



Angle Drive

Mitre gear drive with a 1 to 1 ratio for changing the direction of any shaft through an angle of 90 deg. Driving or driven shaft may be set to work in either horizontal or vertical position. Completely oil tight in any position, it may be used with universal joint to obtain



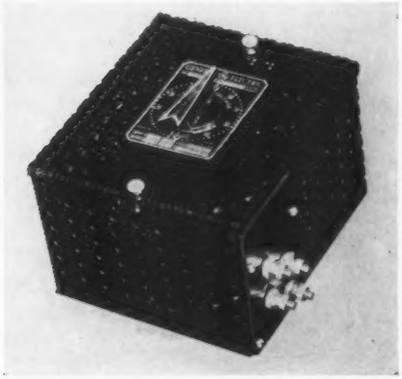
practically any angle of drive. Series ADS transmits 5 hp. at 600 r.p.m., and Series ADH 5 hp. at 1,200 r.p.m. Projecting shafts of 1½ in. dia. may be made in any length. Bearings are of bronze, lubricated with Alemite fittings, on slow-speed units. Anti-friction ball or roller bearings are in the high-speed series. Weighing 50 lb., the unit measures 6½ x 7½ x 8¼ in. Payne Dean & Co., Peekskill, N. Y.

Anti-Corrosive Wrapping

High-strength, pliable and moisture-vapor proof paper is now produced in forms such as rolls, sheets, or bags, for the shipment of parts subject to corrosion. Such wrappings not only seal against atmospheric conditions but also prevent grease, oil or wax from penetrating through from the inside. Riegel Paper Corp., 342 Madison Ave., New York, N. Y.

Adjustable Voltage Auto Transformers

For the adjustable control of uninterrupted voltage and of small amounts of power. Variable-voltage auto transformers in this series are light in weight and small in size. They are designed for panel and bench mounting but can be supplied without the inclosing cases. Instruments are said to have high efficiency and excellent regulation throughout the entire range from no load to



full load. They operate on low input power and low exciting current. The dial on the cover is calibrated to indicate output voltages from 0 to 100 percent of maximum. General Electric Co., Schenectady, N. Y.

Electric Counter

By turning knob pointers of this electric counter to desired digits, any number from 1 to 9,999 can be set so that a control and signal circuit is closed or opened after a desired interval. The circuit may be used to sound an alarm or operate a relay to perform any desired function. Counting ceases at the

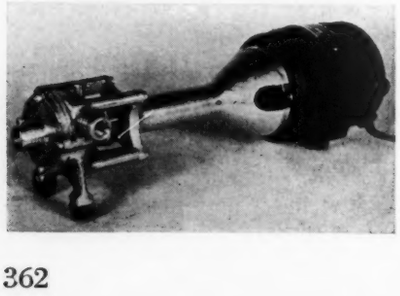
preset number until a reset lever is depressed, making the instrument ready for a new cycle. Called the Monitor Model PDC-E, the unit may be located at a distance from or near to the elements being counted. It is actuated by any switch, relay or photo-electric unit with a closed period of 0.035 sec. or more, and an open period of 0.040 sec. or more. Production Instrument Co., 702–10 W. Jackson Blvd., Chicago, Ill.

Sensitive Bimetal

Through the use of new alloys, a thermostatic bimetal recently developed is said to have increased sensitivity over previous standard types. In addition, reductions in weight and mass are claimed. W. M. Chace Co., Detroit, Mich.

Hot-Liquid Pump

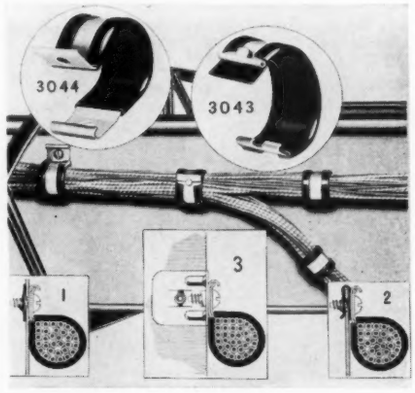
Midget pump suitable for handling hot liquids up to 500 deg. F. Designated Model EH, of centrifugal type,



measuring 12x4x4½ in. Its 1/20 hp. motor, for operation on 115 volts, a.c. or d.c., is coupled to the pump through a 5-in. heat radiating tube. The shaft is supported by outboard ball bearings. Stuffing box is adjusted by means of an easily accessible adjustment ring. Maximum pressure delivered is 21 lb. per sq.in. and maximum volume 7½ gal. per min. These pumps can be furnished in stainless steel, Monel metal, chromium plated bronze, or Hastelloy "C." Eastern Engineering Co., 45 Fox St., New Haven, Conn.

Clamps for Aircraft Wire Installation

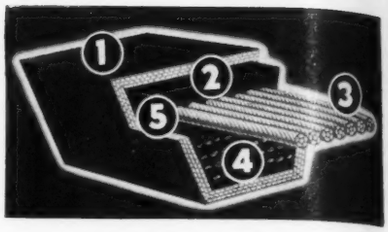
Two series of "Harness Speed Clamps" for the attachment of wire "bundles" in aircraft assembly. Both types have a latching means whereby wires are held together during subassembly operations and during transport to final assembly location. No. 3043 series clamps are used at intermediate locations and where groups of wires branch from the main bundle; No. 3044 are supporting clamps for attachment to structures. Both types come in ten different sizes to accommodate bundles



of wire in group diameters from ¼ to 1½ in. Clamps are covered with plastic channels which are said to have improved abrasion resistance. Three methods of attachment with Speed nuts shown are: (1) with standard, flat type nut; (2) with U type nut snapped over flange edge; and (3) with bracket type nut for attachment at various angles. Tinnerman Products, Inc., 2041 Fulton Rd., Cleveland, Ohio.

V-Belt

V-belt with a new black cover is said to be tougher, insuring longer wear, more uniform pull, and higher resistance to oil and heat. Called the "E-C cord multi-V belt," the belt is made in



all standard cross sections and lengths and sold in sets matched under operating tensions. No. 1 is the cover, No. 2 the tension section, No. 3 the neutral plane with endless, load-carrying compass cords, No. 4 the compression section and No. 5 the insulating gum. Goodyear Tire & Rubber Co., Wilson & Winning Ave., Akron, Ohio.

Totally-Inclosed Motor

"Cowl-cooled" is a new fan-cooled squirrel-cage motor, developed for use in atmospheres containing destructive fumes, abrasive dust, oil or dirt. Ventilating air is blown over external ribs running the length of the frame. In standard N.E.M.A. frame sizes up to 20 hp., motors are available with either ball or sleeve bearings, the latter especially for use where delicate balance and freedom from vibration are first considerations. Ratings are 220, 440 and 550 volts, three-phase, and 220 and 440 volts, two-phase. On 60 cycles, hp. ratings run from ¾ to 25, and on 25 cycles from ½ to 10. Crocker-Wheeler Electric Mfg. Co., Ampere, N. J.

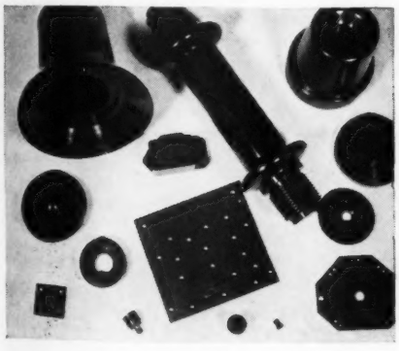
Insulation Tubing

Resistance to brittleness down to minus 15 deg. C. is a feature of a new transparent tubing made especially to secure continued, effective installation on high flying aircraft. Manufacturers state that its toughness and rubber-like qualities make it useful for a variety of industrial and electrical applications. Its transparency allows quick location of wire beaks and identification of wires which have been snaked through it. Transflex is a flexible fibronized tubing available from size No. 14 to ⅝ in. in-

side diameter. Its tensile strength is 3,000 lb. per sq.in., dielectric strength is 850 volts per mil when dry and 815 volts per mil when wet, water absorption is 4 percent in weight after 24 hr. immersion, is not attacked in Wemco oil test (48 hr. at 100 deg. C.), allowable continuous operating temperature is 150 deg. F. Fibron Div., Irvington Varnish & Insulator Co., Irvington, N. J.

Electrical Insulation

A pure synthetic resin containing no filler, "Dilectene," an insulating material, is highly resistant to moisture and electrically stable. It is available in two grades—Dilectene 100, especially for use where resistance to moisture and tendency to resist cold flow are important, and Dilectene 160, where ease of working is the prime factor. The material is supplied in sheets and rods but not tubes. Sheets, approximately 15x33 in., are made in thicknesses from 1/16 to 1 in. Rods of Dilectene are available



up to 1-in. diameter; those of Dilectene 160, up to 2-in. Material can be machined or formed in molds, but it is not recommended for stamping because of its brittleness. Dilectene is recommended for u-h-f applications in radio and television, stator insulation, terminal blocks, strips and boards, standoff insulators, antenna applications, and for electrical equipment in general. Continental Diamond Fibre Co., Newark, Del.

Elapsed Time Meter

With six counter units, elapsed time meter consists essentially of a 600 r.p.m. synchronous driving motor, a gear train, and six numbered wheels. Life-time

bearing lubrication is provided by oil storage reservoir. In use, the meter is connected in parallel with the apparatus on which total time of operation is to be measured. Resetting is accomplished by removing the case and disengaging the number wheel shaft. Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

Manganese Steel Welding Rod

To replace nickel-manganese steel electrodes, an electrode containing 12 to 14 percent manganese, molybdenum and other elements has been developed. Called "V-mang rod," it is said to have ductility and tensile strength equal to or better than nickel-manganese steel rod as applied. Recommended uses are for the repair of fractures in manganese steel parts and for build-up work, depositing a uniform bead similar to that provided by nickel-manganese electrodes. Manufacturers state that it does not replace "Mo-mang," a high manganese, high carbon, molybdenum rod that is not as ductile as the new rod and recommended for build-up work only. V-mang is available bare and coated, in various diameters and 18 in. length. American Manganese Steel Div., American Brake Shoe & Foundry Co., Chicago Heights, Ill.

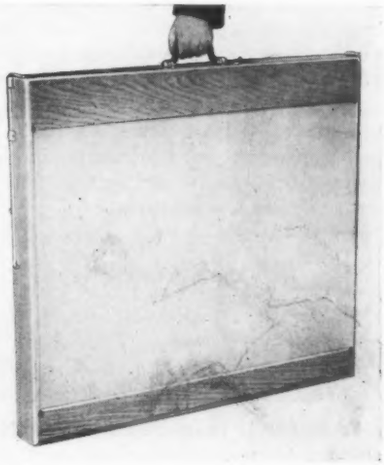
Copying Machine

Continuous-type copier makes contact prints from originals up to 44 in. wide by any length. Tracings may be reproduced from blueprints, black and white prints, or colored originals. The machine will copy originals on either transparent or opaque materials, printed on either one side or two sides. Known as the Revolute No. 8F copier, the machine uses a principle of revolving contact to prevent slipping and blurred prints. A pyrex glass cylinder of 8 in. dia. revolves with original and sensitized material in contact. Inside of the cylinder are located three fluorescent lamps, two white and one gold. White lamps are used in making reproductions from black and white originals, the gold lamp when reproducing from

blueprints. Transparencies printed on one side only are reproduced by transmitted light. Opaque originals printed on two sides are copied by reflected light. With the new photographic copying materials it is not necessary to have a special dark room. Paragon-Revolute Corp., 77 South Ave., Rochester, N. Y.

Portable Tracing Board

With a 24 x 36-in. plate glass tracing surface, the No. 188 portable tracing board is furnished with fluorescent lamps. For use on inclined surfaces, adjustable stop rods hold the board in place. A black leather handle is attached to the top edge. The board is for use with 110-volt, a.c. only. Hamilton Mfg. Co., Two Rivers, Wis.



Books and Bulletins

Handbook of Mechanical Design

GEORGE F. NORDENHOLT, JOSEPH KERR AND JOHN SASSO—*277 pages, 8¾x 11 in. Red Buckram covers. Published by McGraw-Hill Book Co., 330 W. 42nd St., New York, N. Y. Price* $4.

Here is a completely new design handbook that has no parallel among existing references. Purpose is to place before product engineers practical mechanical design methods and procedures now in use in progressive engineering departments, and to do this in a way most convenient to the engineer seeking ideas and data useful in working out design details.

Most of the data is in the form of engineering sketches, charts, tables, formulas and nomographs grouped according to basic design problems. For instance, it includes 36 consecutive pages of tables comparing properties, compositions and applications of all the commonly used engineering materials; 23 pages of sketches showing typical latches, locks and fastenings; and 34 pages of sketches showing details of power transmission elements and mechanisms. Practically no explanations of theory are included. Large-size pages place a maximum of information before the engineer at one glance.

The book has been prepared by members of the PRODUCT ENGINEERING staff in answer to demands for a compilation in book form of design data similar to the Reference Book Sheets and mechanisms sketch pages published regularly in the magazine. Much of the material in the handbook has appeared in past numbers of PRODUCT ENGINEERING, but is here condensed, rearranged and brought up-to-date. Subjects covered in the eight chapters include: charts and tables for general arithmetical calculations; materials; beams and structures; latches, locks and fastenings; springs; power transmission elements; drives and controls; and design data on production methods.

Wood Technology

HARRY D. TIEMAN—*316 pages, 6x9 in. Clothboard covers. Published by Pitman Publishing Corp., 2 West 45th St., New York, N. Y. Price* $3.50.

A comprehensive treatment on the subject of wood technology covering anatomy, identification, description and properties of species and processing.

Chapters that are of more than general interest are those dealing with chemical utilization of cellulosic fibers and synthetic substitutes, paper and pulp from wood, shrinkage and swelling, collapse, drying, gluing methods, mechanical properties, density, absorption, destructive agencies and preservative measures.

An important part of the book is an identification key based on the distinguishing features of American woods.

Manufacturing Processes

MYRON L. BEGEMAN—*579 pages, 6x9 in. Blue clothboard covers. Published by John Wiley & Sons, Inc., 440 Fourth Ave., New York, N. Y. Price* $4.50.

An excellent reference, copiously illustrated, this work will serve either as a students' textbook (as intended) or as a well of information in the engineer's library. It presents, in layman's language, fundamentals of principal manufacturing processes, materials and machinery. Photographs, sketches and tables are well chosen and effectively placed.

Eight chapters cover in a comprehensive manner such subjects as foundry practices, pattern making and applications, casting, plastic molding, heat treating of steel, welding, soldering, hard surfacing, and hot and cold forming of metals. One section deals with inspection and gages, one with grinding wheels, machines and abrasives and another with threads and thread cutting.

The remaining nine chapters are devoted to functions and operation of production machinery—lathes, lathe tools, shapers, planers, drilling and boring machines, milling, cutting, sawing and broaching machines and tools. Appendices include the S.A.E. numbering system, S.A.E. steels, and suggested laboratory projects.

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Surface Hardness of Metals

BRUCE CHALMERS—*5¾x8¾ in., 19 pages. Publication No. 108, Tin Research Institute. Distributed by Battelle Memorial Institute, 505 King Ave., Columbus, Ohio.*

Method is described for measuring the hardness of thin surface layers of metals and other materials, in which the reduction in optical reflectivity is determined when known quantities of sand are allowed to impinge on the surface under standardized conditions.

Aluminum Alloy Properties

Aluminum Research Laboratories Technical Paper No. 6. Paper covers, 20 pages, 8x10½ in. Published by Aluminum Company of America.

Typical stress-strain curves, and stress-modulus curves derived therefrom, are presented for some aluminum alloy products of particular interest to the aircraft industry. These products are 24S-T sheet, tubing, and extruded shapes, Alclad 24S-T sheet, 24S-RT sheet, and Alclad 24S-RT sheet. Curves are given for both tension and compression, and in the case of sheet, for two directions, parallel and normal to the direction of rolling.

Effect of Range of Stress on the Fatigue Strength of Metals

JAMES O. SMITH—*6x9 in. 52 pages, paper covers. Engineering Experiment Station Bulletin Series No. 334, University of Illinois, Urbana, Ill. Price 55 Cents.*

A comprehensive study of test data to determine the manner in which the range of stress effects the fatigue strength of metals. Data studied involved ranges of repeated axial stress from tension to tension, tension to compression, compression to compression, and of repeated torsional shearing stress, in both ductile and brittle metals for specimens with and without notches. The author's main purpose is to extend the development of the general law or laws that control this phenomena over a wide field of applications.

Luminescent Coatings

Federation of Paint and Varnish Production Clubs, Report of Sub-Committee No. 43, Project No. 6, New York Paint and Varnish Production Club. Paper covers, 10 pages, 6x9 in.

Luminescent coatings are described from the standpoint of their properties and uses to indicate their practical value and limitations. The booklet is intended particularly for manufacturers of such coatings. Photoluminiscent coatings only are considered. Those containing radio-active materials are not included. Publication was prompted by the importance of such coatings in war activities.

Stainless Steels

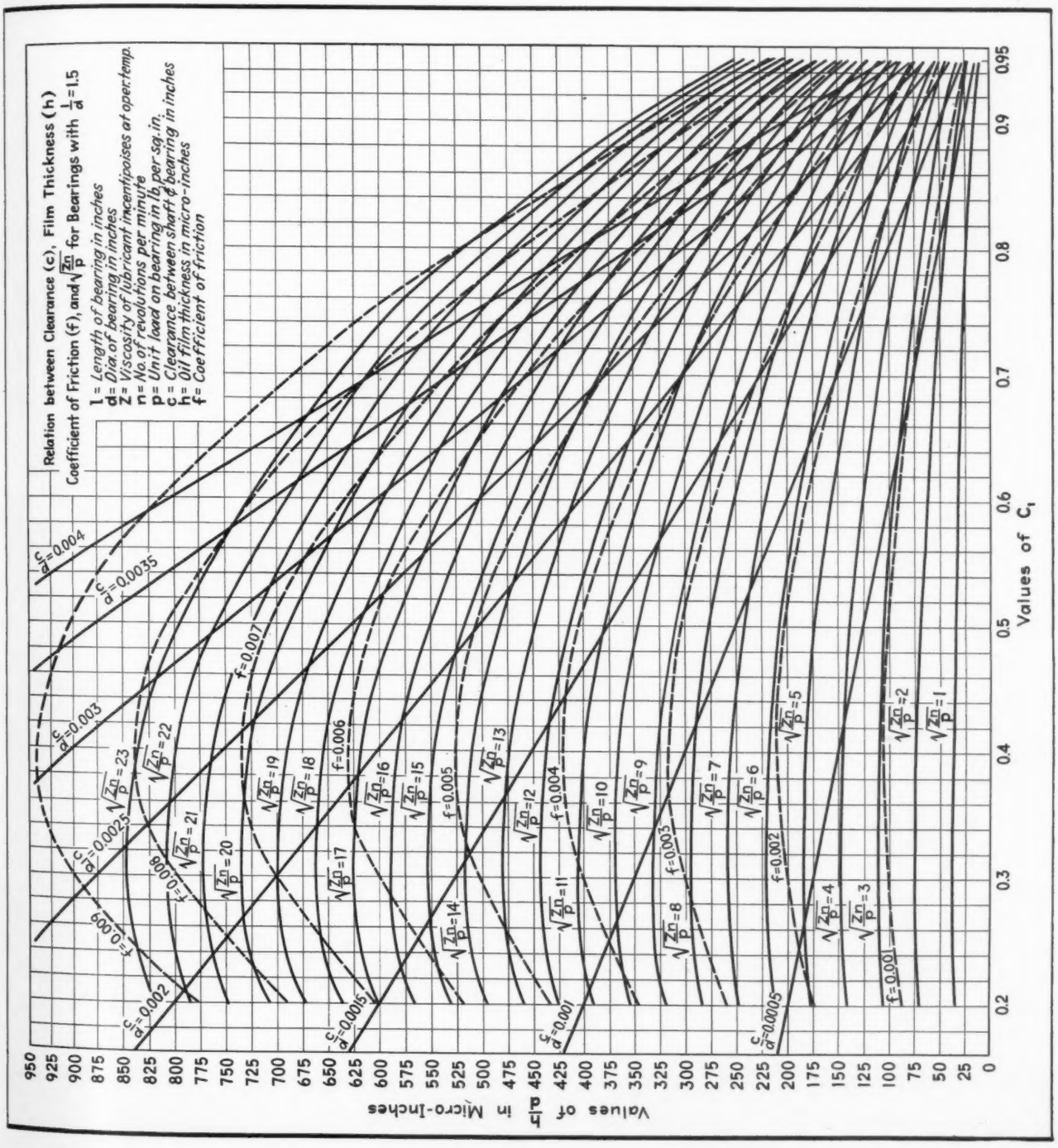
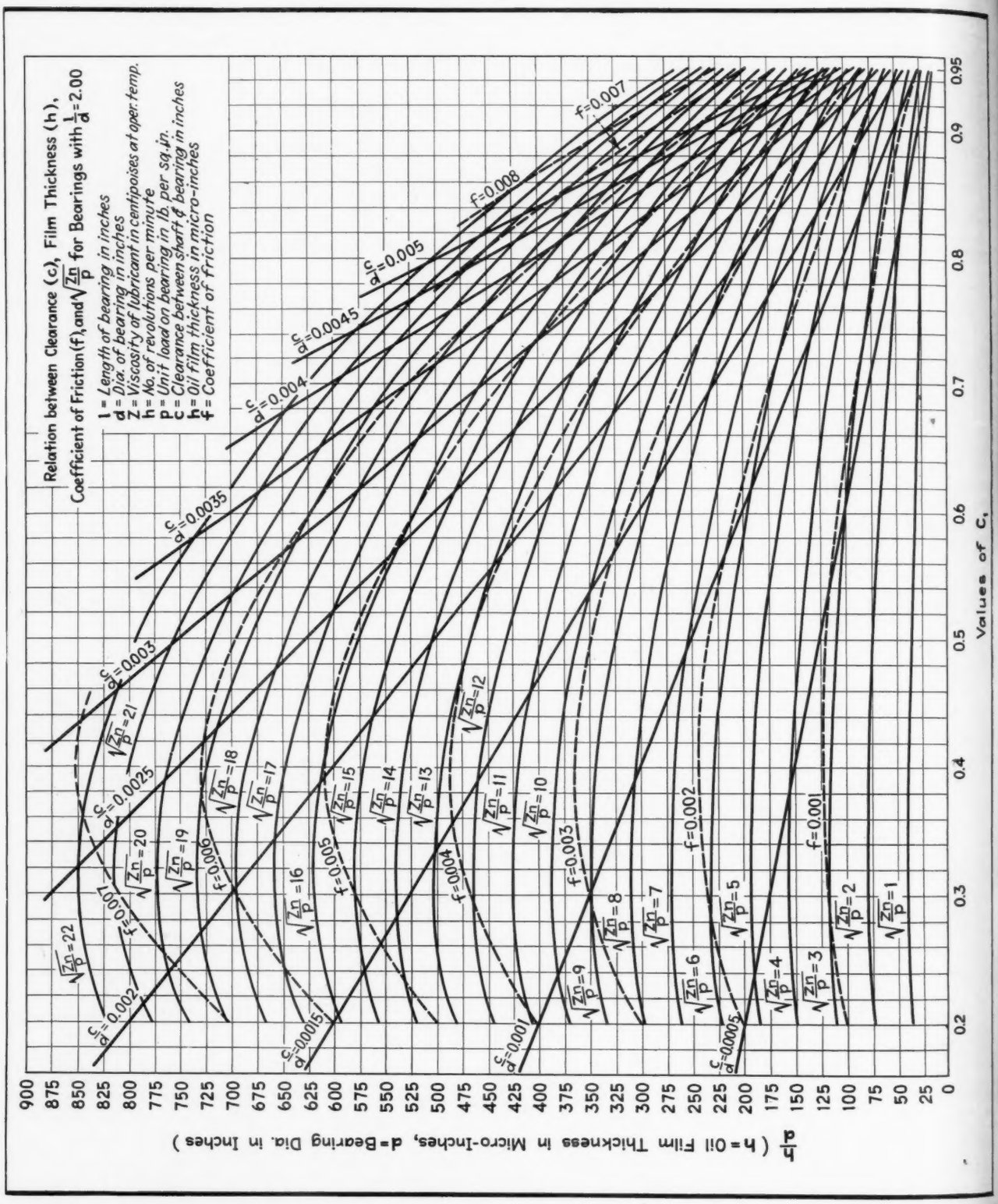
32 pages, 8½x11 in. Published by Allegheny Ludlum Steel Corporation, Pittsburgh.

An elementary discussion of stainless steel, this booklet begins with a general description of this large group of alloys along with tables of comparable properties. Corrosion resistance is presented separately in tabular form. The major portion of the booklet is on fabrication and deals briefly with welding, drawing and blanking, machining, spinning, upsetting and forging, riveting, shearing, soldering and brazing, annealing and heat treatment, grinding and polishing, and surface treatment.

SLEEVE BEARING DATA

BRUNO SACHS, *Technical Director, Johnson Bronze Company*

CHARTS on this and the next page are supplements to the article "Sleeve Bearings," see Part I p. 248, May, and Part II p. 322, June, PRODUCT ENGINEERING. They show relation between clearance, film thickness, coefficient of friction and other factors for bearings having a length diameter ratio equal to 1.5, and equal to 2 respectively. Similar data for bearings having a length diameter ratio equal to one is shown on p. 252. May P.E.


Relation between Clearance (c), Film Thickness (h)
Coefficient of Friction (f), and $\sqrt{\frac{Zn}{p}}$ for Bearings with $\frac{l}{d}$=1.5
l = Length of bearing in inches
d = Dia. of bearing in inches
Z = Viscosity of lubricant incentipoises at oper.temp.
n = No. of revolutions per minute
p = Unit load on bearing in lb. per sq.in.
c = Clearance between shaft & bearing in inches
h = Oil film thickness in micro-inches
f = Coefficient of friction
Values of Clr in Micro-Inches
Values of C_1

SLEEVE BEARING DATA (*continued*)